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Technical Assessment of Advanced Cooling Technologies in the Current Market

by
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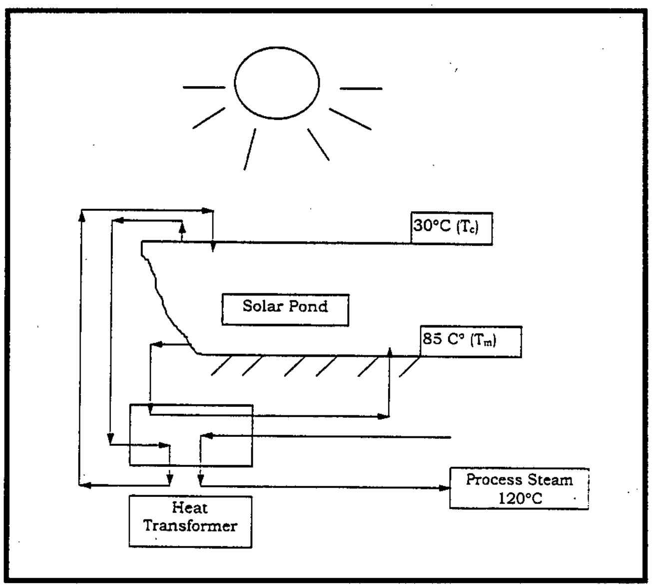

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Foreword

This study was conducted for the Directorate of Military Programs, Headquarters, U.S. Army Corps of Engineers (HQUSACE), under Project 40162784T45, "Advanced Cooling Technologies"; Work Unit AT45-UL-X27, "Advanced Cooling Technologies." The technical monitor was Tim Gordon, CEMP-ET.

The work was performed by the Utilities Division (UL-U) of the Utilities and Industrial Operations Laboratory (UL), U.S. Army Construction Engineering Research Laboratories (USACERL). The USACERL Principal Investigator was Dr. Chang W. Sohn. Part of this work was done by the University of Illinois at Urbana-Champaign, Urbana, IL 61801 under contract No. DACA88-94-D-0011-06. Martin J. Savoie is Chief, CECER-UL-U, and Dr. John T. Bandy is Operations Chief, CECER-UL. The associated Technical Director was Gary W. Schanche is, CECER-UL. The USACERL technical editor was William J. Wolfe, Technical Resources.

COL James A. Walter is Commander and Dr. Michael J. O'Connor is Director of USACERL.

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1 Introduction

1.1 Background

A variety of cooling and heat pump technologies is available in today's market. These include motor- or engine-driven vapor-compression systems, steam powered or direct-fired absorption systems, and novel systems such as desiccant systems, thermoelectric cooling, and systems that use geothermal heat sources and sinks. A detailed examination of the state of technology in advanced cooling systems—in terms of system efficiencies and first costs—can be used in determining the life cycle cost (LCC) of the system for a given military installation, based on life of the system and the discount rate. Comparison of the LCCs of different systems on a net present value (NPV) basis can be used as a selection criteria that minimize overall system costs. The NPV of a system includes the first cost (purchase cost) and LCC. The operating costs include yearly maintenance and energy costs. The latter were found to have a greater effect on yearly expenditure than the former. Such a study will also have significant input to the plan toward revitalization of military facilities as cooling facilities at Army installations are renewed or replaced (Levins and Ternes 1994).

1.2 Objective

The objective of this study was to review cooling technologies available in the current market and to promote the introduction of advanced cooling technologies to Army facilities. The extensive cost information compiled in this report will serve as handy reference for the Army engineers in selecting the most cost-effective cooling systems for installation-specific applications. The theoretical and factory catalogue efficiencies in this report will also serve as baseline data for comparison to the field performance data.

1.3 Approach

A survey and a review of cooling system manufacturers as well as technical papers and articles was conducted. Detailed exploration was made into aspects and equipment that was been the subject of sufficient attention in recent studies (EPRI 1992; WGNAS 1993; SAI 1995; Cler 1995). An examination of ideal and practical system efficiencies in terms of the coefficient of performance (COP) was also conducted. These are compared to the rated system efficiencies quoted by the manufacturers. The COP was used in estimating the yearly energy costs. The main types of systems examined were vapor-compression and absorption systems. Heat pump and desiccant systems were also surveyed.

The survey of manufacturers included information on type of refrigerant, type of compressor, ton rating, drive system, system first cost, rated COP, and evaporator and condenser temperatures. Significant references are the inlet temperature for the respective source or sink (i.e., T_e is the chilled water inlet temperature, and T_c is the ambient air or cooling water inlet temperature). As will be noticed, manufacturers use various T_c and T_e for rating the COP of their units. A method for converting the COPs at various T_c and T_e to a common reference is presented in the study to allow for comparison of units produced by various manufacturers. Table 1 lists a summary matrix of the systems. $(COP_c)_e$ denotes COP for cooling based on shaft power or electric drive; $(COP_c)_s$ denotes COP based on a heat or fuel source. Detailed versions of the matrix are included in later sections. A vigorous effort was made to acquire prices of chillers from various manufacturers. Some were kind enough to comply with a promise of anonymity. Others simply did not respond to the request. One underlying reason was that, in the case of large units (500 to 1,000 tons), sales tax paid by commercial customers differs between special order and standard equipment (as evidenced by a price list).

The electric motor-driven vapor compression system was selected as a baseline cooling system for comparison to other systems. Catalogue performance of each cooling system was collected through contacts with the system manufacturers. Based on the first cost obtained in the market and the catalogue performance of the systems, life cycle cost analysis was performed for a number of selected cooling technologies. Corroboration of the theoretical and field performance of natural gas cooling and GHP systems will be made in the near future when the field performance data from the demonstrations under the Strategic Environmental Research and Development Program (SERDP) and the Federal Energy Management Program (FEMP) become available.

Table 1. Summary matrix.

Mfr. Code	Capacity (Tons)	Compressor	$(COP_c)_e$	$/Ton	Condenser Cooling
(a) Motor driven vapor-compression systems.					
B	200-7000	Centrifugal	5.6-6.1		Water
P	200-1400	Centrifugal	6.4	218-417	Water
R	150-2100	Centrifugal	5.4-5.8	172-482	Water
B	150-350	Screw	5.4	150-350	Water
C	50-900	Screw	4.4	162-360	Water
P	70-400	Screw	2.7-2.9	335-465	Air
P	70-450	Screw	3.5-5.8	200-423	Water
B	15-280	Recip.	3.0-3.3		Air
B	40-225	Recip.	3.5-4.3		Water
R	50-450	Recip.		349-500	Air
R	60-250	Recip.		228-367	Water
P	10-60	Scroll	2.9-3.6	444-757	Air
P	20-60	Scroll	3.8-4.2	451-685	Water
S	1	Rotary	2.8	559	Air
T	1	Rotary	2.6	519	Air
(b) Engine driven vapor-compression systems.					
D	2000-6000	Centrifugal			Water
A	250-1040	Screw	1.32		Water
D	250-1100	Screw	1.57-1.81	513-760	Water
L	50-4000	Screw	1.0-2.0		Water
M	110-700	Screw	0.85-1.3	500-722	Water
A	50-300	Recip.	1.38-1.52		Water
B	25	Recip.	1.0		Air
D	95-315	Recip.		635-1000	Water
L	50-300	Recip.	1.0-2.0		Water
M	60-75	Recip.	0.95-1.3	917-1000	Water
N	15-25	Recip.	0.80	1088-1462	Air
R	3-4	Recip.	0.90	2000-2333	Air
(c) Single-effect absorption systems.					
B	108-680	Steam	0.7	236-601	Water
P	112-1660	Steam	0.68	272-795	Water
Q	5-10	Steam	0.7	2326-4640	Water
R	120-1377	Steam	0.69	205-590	Water
*H	3-5	D.F.**	0.48-0.62		Air
(d) Double-effect absorption systems.					
B	135-1000	D.F.	0.97	415-933	Water
B	100-1700	Steam	1.2	365-1104	Water
F	20-1500	D.F.	0.95-1.0	500-1700	Water
F	100-1500	Steam	1.4	400-900	Water
P	100-1100	D.F.	1.0	604-1320	Water
P	385-1125	Steam	1.2	446-683	Water
Q	30-100	D.F.	0.95-1.0	718-953	Water
R	200-1000	D.F.	0.92	516-671	Water
R	440-1500	Steam	1.16-1.19	416-493	Water

* Systems use Ammonia/Water, all others use Li-Br/Water
**D.F. are direct fired systems

1.4 Mode of Technology Transfer

The information in the report will be delivered to Army engineers through professional meetings (such as the U.S. Army Corps of Engineers [USACE] Mechanical and Electrical Engineering Training Conference and Department of Defense [DOD] Energy Managers Meeting). Portions of this report will be included in a Technical Bulletin for installation Directorate of Public Works (DPW) staff. The system performance data will also be used to update the database in the Renewables and Energy Efficiency Planning (REEP) Program and in the revision of Technical Manual (TM) 5-810-1, *Mechanical Design Heating, Ventilating, and Air Conditioning*.

1.5 Metric Conversion Factors

The following metric conversion factors are provided for standard units of measure used throughout this report:

1 in. = 25.4 mm	1 lb = 0.453 kg
1 ft = 0.305 m	1 gal = 3.78 L
1 sq ft = 0.093 m^2	1 ton (refrigeration) = 3.516 kW
1 cu ft = 0.028 m^3	1 BTU = 1.055 kJ

2 Definition of Performance Parameters

This chapter describes the performance parameters that have been used in the evaluation of various systems. The main performance parameter of concern for this study is the system COP. The COP limits for ideal systems were developed as well as the COP limits for real cycles and systems. These can give an indication of the theoretical and practical maximum COP values for various systems and can be compared to manufacturer rated COP information. The effect of T_c and T_e on COP is also discussed. Desiccant systems are introduced and their use and method of rating are investigated. Information on novel systems such as heat pumps, thermoelectric units, and others are presented for possible consideration.

The coefficient of performance for cooling based on shaft work or a heat or fuel source, and those for a heat pump based on shaft work or a heat source are defined in the following manner. The coefficient of performance for refrigeration based on shaft work to remove heat at a temperature T_c is:

$$\left(COP_c\right)_e = Q_c / W_{in} \qquad \textbf{Eq 1}$$

where Q_c is the heat removed and W_{in} is the shaft work that can be supplied by an electric motor or engine. In many systems, the cold body takes the form of chilled water. Some manufacturers provide information on the power consumption per ton of refrigeration. This can be related to the $(COP_c)_e$ by:

$$kW / ton = 3.516 / \left(COP_c\right)_e \qquad \textbf{Eq 2}$$

Another performance measure sometimes cited in manufacturers literature is the energy efficiency ratio (EER in Btu/W). This can be converted to a COP value by:

$$\left(COP_c\right)_e = (EER) / 3.413 \qquad \textbf{Eq 3}$$

The coefficient of performance for a refrigeration system based on a heat source (either gas fired or steam) is:

$$\left(COP_c\right)_e = Q_c / Q_s \qquad \textbf{Eq 4}$$

where Q_c is the heat removed and Q_s is the heat supplied by the heat source. This definition is directly applicable to absorption type systems. The matrix in Table 1 lists the rated COP quoted by manufacturers of various systems. The vapor-compression systems use $(COP_c)_e$ and the absorption systems use $(COP_c)_s$ as defined above. The two COPs can be related by the efficiency of the power generation facility defined as η_e, where:

$$\eta_e = W_{out} / Q_s \qquad \textbf{Eq 5}$$

We can see that:

$$(COP_c) = (Q_c / W_{in})(W_{out} / Q_s) = (COP_c)_e(\eta_e) \qquad \textbf{Eq 6}$$

in which η_e is based on the ambient temperature in the refrigeration mode. For a system driven by shaft work that can act as a heat pump, the coefficient of performance in the heating mode can be determined by:

$$(COP_h)_s = Q_h / Q_s \qquad \textbf{Eq 7}$$

where Q_h is the heat supplied and W_{in} is as previously defined. This formula can be reduced to the following form:

$$(COP_h)_e = (COP_c)_e + 1 \qquad \textbf{Eq 8}$$

Similarly, the COP of a heat pump with the engine driven by a heat supply can be defined as:

$$(COP_h)_s = Q_h / Q_s \qquad \textbf{Eq 9}$$

where Q_h is the heat supplied to the conditioned space and Q_s is the heat provided by the heat source. This relation can then be manipulated to the form:

$$(COP_h)_s = (COP_h)_e(\eta_e^{'}) = (COP_c)_s + \eta_e^{'} \qquad \textbf{Eq 10}$$

where η_e' is the engine efficiency based on the ambient temperature in the heat pump mode. It is expected that $\eta_e' > \eta_e$ since a heat engine would operate at an improved efficiency at a reduced ambient temperature. This relation is discussed further in Section 2.4. For a heat pump based on an absorption system using steam or a direct-fired heat source, the coefficient of performance can be defined by:

$$(COP_h)_s = Q_h / Q_s \qquad \textbf{Eq 11}$$

The above relation can be manipulated to the form:

$$(COP_h)_s = (COP_c)_s + 1 \qquad \text{Eq 12}$$

due to the use of the heat rejected in both the condenser and the absorber. Absorption systems are discussed in detail in Chapter 5.

2.1 Ideal Cycle

The ideal cycle can be described by four thermodynamic processes. First, there is a reversible isothermal process in which heat is transferred to a working fluid from a high temperature reservoir. Second, there is a reversible adiabatic process in which the temperature of the working fluid decreases from the high temperature to the low temperature. Third, there is a reversible isothermal process in which heat is transferred to a low temperature reservoir. Fourth, there is a reversible adiabatic process in which the temperature of the working fluid increases from the low temperature to the high temperature (Sonntag and Van Wylen 1982). The above four processes describe the Carnot cycle, which converts heat into work. This cycle provides the highest level of performance for an engine driven by a heat input from a reservoir. Since all processes in the cycle are reversible, the cycle can be run "backwards" as a refrigerator. The ideal performance of a refrigerator or heat pump based on a heat input is obtained by a Carnot engine driving a Carnot refrigerator of identical work output and work input. For a refrigeration cycle, the performance is measured by the coefficient of performance (COP), which is the amount of heat removed from the low temperature reservoir divided by the net work input. For the Carnot refrigeration system described, the performance can be defined as:

$$(COP_c)_s = T_c(T_s - T_o) / T_s(T_o - T_c) \qquad \text{Eq 13}$$

in which $(COP_c)_s$ is the coefficient of performance for cooling based on a heat input, T_c is the chilled water temperature, T_s is the temperature of the heat source that drives the system, and T_o is the temperature of the surroundings or heat sink (for the condenser of a refrigerator) or source (for the evaporator of a heat pump). All the temperatures are in Rankine. For a heat pump, this ultimate performance is given by the relation:

$$(COP_h)_s = T_h(T_s - T_o) / T_s(T_h - T_o) \qquad \text{Eq 14}$$

In this relation $(COP_h)_s$ is the coefficient of performance for heating based on a heat source, T_s and T_o are as described for Eq. 13, and T_h is the temperature at which heat is delivered by the heat pump. For example, giving T_s = 2,000 R, T_o = 500 R, and T_h = 550 R, $(COP_h)_s$ = 8.25, i.e., for one Btu applied at 2,000 R, 8.25 Btu would be available at 550 R or 90 °F. The above COPs are the theoretical maximum possible for any device, but this level of performance cannot be approached by a practical system with real working fluid. However, the above relations, especially for $(COP_h)_s$, indicate that heat supplied at a high temperature or chemical potential is worth more than the same amount of heat supplied at a low temperature, such as waste heat.

2.2 Real Cycle

Practical refrigeration systems use the standard vapor-compression cycle to carry out the refrigeration function. The simple ideal vapor-compression cycle offers a practical estimate of the capabilities of a refrigeration system. In the vapor-compression cycle, a saturated liquid at condenser temperature T_c is throttled to the evaporator pressure where the refrigerant removes heat from the low temperature body (typically chilled water) by evaporation at T_e. The saturated vapor from the evaporator undergoes an adiabatic (isentropic) compression process from the evaporator pressure to the condenser pressure. The compressed and superheated vapor rejects heat to the warm body (ambient air or water) and condenses to a saturated liquid at T_c, and the cycle repeats. The main deviation of the vapor-compression cycle from the Carnot cycle is that the compression leads to a superheated vapor and that the throttling process is irreversible. It is virtually impossible to compress, in a practical device, a liquid-vapor mixture. The use of saturated vapor in the compression process suits the vapor-compression cycle to practical refrigeration applications. The compressor can be driven by either an electric motor or a heat engine. Due to the inefficiencies of the compressor, irreversibilities, and other losses in the system, the COP for the ideal vapor-compression cycle tends to be far below that of the Carnot cycle. The actual vapor-compression cycle deviates from the ideal primarily due to the adiabatic efficiency of the compressor, the pressure drops associated with fluid flow, and heat transfer to the surroundings (Sonntag and Van Wylen 1982). The actual COP of the cycle depends on the values of T_c and T_e, as well as the type of refrigerant used.

2.3 Desiccant Systems and Dehumidification

Two primary objectives of air-conditioning systems are to reduce the humidity of an air stream (latent cooling) and to lower the temperature of an air stream (sensible cooling). Many alternative technologies exist to achieve dehumidification. There are three basic methods to remove moisture from air (Munters 1990). One method is to cool the air to the dew point temperature. The dew point temperature is the temperature at which moisture will condense out of the air onto any nearby surface. A second method is to increase the total pressure, which also causes condensation. Since the ambient pressure in HVAC applications is constant, this method does not apply. A third method is to use a desiccant material, which will attract moisture. Specific humidity is the amount of moisture in the air per unit mass of air. A typical unit is grains of water per pound of dry air. (There are 7,000 grains in a pound.)

The most common means of achieving dehumidification is by cooling that uses the vapor-compression cycle. Three basic equipment configurations for dehumidification (Munters 1990) are: (1) direct expansion cooling, (2) chilled liquid cooling, and (3) dehumidification-reheat systems. Direct expansion (DX) systems are characterized by a refrigerant gas expanding directly into an air cooling coil. Heat is removed from the air stream at the coil. Chilled liquid systems are characterized by a secondary loop. The refrigerant gas expands into a liquid cooled coil. The liquid is then circulated through a cooling coil to cool the air that is to be dehumidified. The liquid used in the secondary loop is usually either chilled water, glycol, or brine. Dehumidification-reheat systems use either direct expansion or chilled liquid for cooling, but then reheat the air before returning it. Most small commercial dehumidifier units use this configuration. The mechanical arrangement is such that the refrigerant condenser is downstream of the evaporator. This allows the air stream to be reheated to a comfortable level after dehumidification by using the essentially free energy from the condenser coil.

Conventional chiller systems can only achieve dehumidification by cooling the air to lower its humidity by condensation followed by heating to the desired temperature. If the air is cooled to a temperature below 32 °F, it can cause the condensate to freeze. Frost will form on parts of the cooling coil, which reduces the heat transfer because the frost acts as insulation and restricts airflow over the coil. Conservatively, this means that air be cooled to 40 °F as a limit, which corresponds to a lowest humidity of 36 gr/lb dry air.

Desiccant dehumidifiers do not cool the air to condense its moisture. Desiccant dehumidification does not provide cooling, but rather shifts the latent cooling load to a sensible cooling load. Desiccant materials have a low vapor pressure at their surface and attract moisture from the air because the pressure exerted by the water in the air is higher. The pressure gradient causes water molecules to move from the air to the surface of the desiccant at a reduced vapor pressure, resulting in dehumidification of the air. Desiccants can be either solids or liquids and collect moisture due to vapor pressure gradients. Their surface vapor pressures are a function of their temperature and moisture content. However, desiccants react differently to moisture. Adsorbent desiccants collect moisture like a sponge and are usually solid materials. Water is collected on the surface and in the crevices of the material. Absorbent desiccants undergo a chemical or physical change as they collect moisture and are usually liquids or solids that change phase to liquids as they absorb moisture.

Desiccant dehumidification is achieved in three stages. The desiccant material begins dry and cool at a temperature of approximately 50 °F with a low surface vapor pressure. As the desiccant attracts moisture in the process of sorption it becomes moist and warm and the pressure gradient degrades. The temperature of the desiccant rises to about 95 °F. The temperature of the air rises because the removal of moisture results in a release of heat (the heat of sorption). The desiccant vapor pressure approaches that of the surrounding air and is no longer capable of acquiring moisture. The desiccant is then regenerated in the process of desorption by removing it from the moist air stream and heating it. It is then placed in a different air stream such that the desiccant surface vapor pressure is higher than the surrounding air. The pressure gradient causes the moisture to leave the surface of the desiccant material and enter the air stream until the pressure is equalized. The desiccant reaches a temperature of about 250 °F after heating and desorption. The desiccant material is then cooled to approximately 50 °F to achieve the initial low vapor pressure and the cycle repeats (Munters 1990)

Many of the features that make desiccant units appear attractive are actually not applicable in many situations. For example, desiccant units themselves are free of chlorofluorocarbons (CFCs). However, if sensible cooling is required, it will be necessary to use some type of additional refrigeration system that may contain CFCs. The claim that desiccant systems will result in downsizing existing chillers is also not true in many situations for which a moderate degree (greater than 36 gr/lb) of dehumidification is required. Desiccant systems in these applications may require a larger power input than equivalent vapor-compression systems. Desiccant systems do not necessarily have lower annual maintenance costs. Desiccant systems have many more components, such as additional fans, ducts, and a boiler, all of which must be maintained. Solid desiccants will also decay with use and will eventually require replacement. Some solid desiccant material will inevitably be lost in the air stream and swept away. Additional filtration of air will be needed.

A common source of confusion with desiccant systems is the rating of capacity and the measurement of efficiency (American Gas Cooling Center 1994) because drying of air does not necessarily involve cooling. The desiccant cycle is driven by thermal energy. The efficiency of the cycle improves for desiccant materials that have a high moisture capacity and low mass. The ideal desiccant dehumidification system is characterized by an infinite surface area for moisture collection and an infinitely small mass (Munters 1990). These characteristics result from the fact that the required heating and cooling energy is directly proportional to the mass of the desiccant and the mass of the mechanism that presents the desiccant material to the air stream. System capacity can be rated in terms of water removal rate (lb/hr), airflow capacity (cfm), or cooling capacity (tons). However, the moisture removal rate varies depending on input and output conditions. There are currently no industry-accepted standard methods of calculating COP for desiccant dehumidification systems (American Gas Cooling Center 1994). Sources of variance in quoted values of COP include the heat sources used for regeneration, the source of regeneration air, and method of cooling the dehumidified air. Manufacturer quotes of COP should be carefully examined to determine how COP is defined (American Gas Cooling Center 1994). A better method of comparing alternative systems to desiccant refrigeration is to calculate an equivalent cooling load for the desiccant system that achieves the desired level of humidity. This equivalent load can then be used to compare desiccant systems to alternative refrigeration systems.

Figure 1 illustrates the concept of "equivalent load." The energy input into the desiccant system in Figure 1(a) includes 237,000 Btu/hr for the reactivation of the desiccant wheel and 59 tons of refrigeration. Other components include a process air fan, heat pipe heat exchanger, and reactivation fan and motor. A simple vapor-compression refrigeration system in Figure 1(b) can achieve the same air-conditioning as the more complex desiccant system in Figure 1(a). With 2,000 scfm of fresh air and 8,000 scfm of return air, the mixture enters the refrigeration system at 71 °F and 53 gr moisture/lb dry air. This condition corresponds to an enthalpy of 26.2 Btu/lb dry air. Refrigeration of the air to 40 °F saturated at 36 gr/lb dry air, corresponding to an enthalpy of 15.2 Btu/lb, requires only 26.2 tons of refrigeration. The air can be reheated by the heat rejected by the condenser of the chiller to 55 °F at 36 gr/lb. The same process air fan would be required, but the other components of the desiccant system would not be needed. The result is that the simplified vapor-compression system is more energy efficient than the desiccant system. The vapor-compression system most likely costs less in maintenance because there are fewer components.

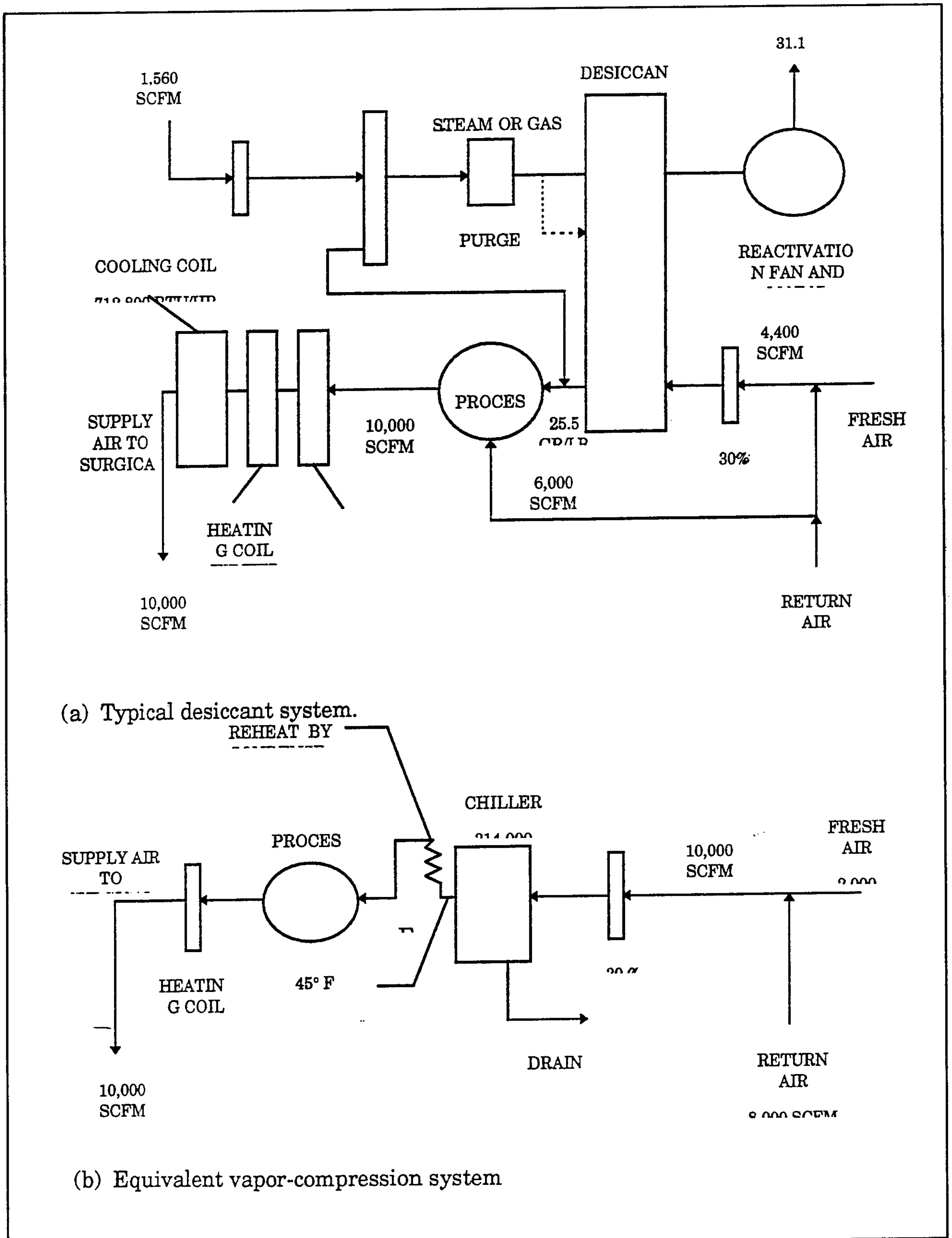

Figure 1. Comparison of desiccant and vapor-compression systems for a hospital operating room.

Results from equivalent cooling load calculations indicate that desiccant systems are not likely to be competitive for air-conditioning applications unless a humidity below 36 gr/lb of dry air is required, or when it is acceptable to achieve dehumidification without cooling. For higher levels of humidity than that, it is more economical to use alternative refrigeration techniques such as simple vapor-compression.

At least five unique equipment configurations exist to achieve desiccant dehumidification (Munters 1990). These include the liquid-spray tower, solid packed tower, rotating horizontal bed, multiple vertical bed, and rotating honeycomb. Each design has both benefits and limitations. Items to consider when evaluating different desiccant systems include installed cost, operating cost, demonstrated operational reliability, and design assumptions (Munters 1990). The cost of the dehumidifier is often less significant than the complete installation cost. The cost of operation and maintenance can be significant. However, many designs take advantage of low cost energy sources that lead to significant operational cost savings. The selection of equipment is entirely dependent on design assumptions, which can differ among engineers and manufacturers for a given application.

Desiccant systems cannot simply be substituted for conventional cooling systems on a ton-for-ton basis, although some equivalence can be drawn as shown in the above and the next example. A typical case is a system for entering air at 102 db/79 wb (h = 43.2 Btu/lb air) at 8,000 cfm (without return air) and leaving air at 84 db/64wb/54 gr/lb. With a natural gas boiler for 26 gpm of water with 600 MBtu/hr input, this is equivalent to 53 kW of electric power (assuming 30 percent generating efficiency) plus a 7.5 hp (5.6 kW) fan for the reactivator, thus a total of 58.6 kW equivalent. A vapor-compression system with cooling to 47 dB at saturation (h = 18.4 Btu/lb) will need 73.5 tons of refrigeration, which, for a COP of 5, will call for a power input of 51.7 kW and reheating by heat rejected from the condenser to give the desired leaving air-condition.

The fact that the desiccant system uses natural gas makes it possible to compare desiccant systems to other technologies on the basis of "gas versus electric power," with the qualification that the desiccant system has many more components than other gas-powered systems. Although manufacturers of desiccant systems tend to prefer ratings in cfm, an equivalent cooling load can be also used as a basis for comparison to other air-conditioning systems that can also control humidity for personnel environment.

For an operating condition of leaving air at below 36 gr/lb moisture, a desiccant system will have to be used, either solid or liquid (R.H. below 1 percent is attainable). When a humidity below 36 gr/lb dry air is needed or when cooling is not required with dehumidification, a desiccant system should be the system of choice.

Many applications are suitable for desiccant dehumidification. Among the major uses are corrosion protection, condensation protection, mold and fungus prevention, product drying, and dry cooling (Munters 1990). Corrosion prevention is a major consideration for the U.S. military. Dry storage is necessary for the long-term protection of inactive ships, machinery, and weapons. Dry conditions also protect computers and other electronic equipment by preventing tiny layers of corrosion to build up on surfaces of circuits. Annual maintenance costs for military equipment can be cut drastically by the use of dry storage achieved by desiccant dehumidification equipment. An 8,000 cfm, gas-fired, desiccant unit was recently installed in a 12,000 sq ft addition to a commissary at Fort Campbell, KY to fight excessive humidity in the freezer area. Humidity levels were lowered and energy savings were projected to be $40,000 annually due to a reduction in the cooling load (American Gas Cooling Center flier 1994).

Several factors should be considered when choosing between desiccant and cooling dehumidification systems. Cooling and desiccant systems are most economical when used together because the technologies complement each other. The proper mix of cooling and desiccant equipment can be partially determined by the difference in cost of electrical power and thermal energy for a given geographic location. In general, cooling-based systems are more economical than desiccant systems at high air temperatures and moisture levels, and when it is desired to dry the air to saturated conditions. Desiccant systems are sole choices when drying air to create low relative humidity corresponding to those below 36 gr/lb of moisture and when treating ventilation air for systems that use ice storage in which no refrigeration is needed (Munters 1990).

2.4 Thermoelectric Cooling and Other Novel Devices.

Thermoelectrics represent a novel method of achieving cooling for specific applications. Thermoelectric modules are small, solid-state heat pumps. A module is composed of an array of n- and p-type semiconductors connected electrically in parallel and thermally in series. The semiconductors are sandwiched between metallized ceramic substrates. The thermoelectric module absorbs heat at one end of the device and rejects heat at the opposite end when the module is connected to a DC power source. This phenomenon is known as the "Peltier effect." If the current flow is reversed, the direction of the heat flow is reversed (Angrist 1971).

Thermoelectric refrigeration for units that generate small amounts of cooling (a fraction of a ton) has several advantages. Ceramic wafers can be designed to fit in a space the size of a fingernail and to weigh as little as one gram. Thermoelectrics can be used in applications that require mobility, such as beverage or medical coolers. Since thermoelectric coolers have no moving parts, they are very reliable. Another advantage is that thermoelectrics provide noiseless operation. Thermoelectrics are environmentally friendly because they use no CFCs or chemical refrigerants.

Thermoelectric refrigeration has several disadvantages. Attainable $(COP_c)_e$ are low for thermoelectric refrigeration. The performance of a thermoelectric cooler depends on the figure of merit of the semiconductor materials composing the p-n junctions. Typical n-type compounds are 75 percent Bi_2Te_3 and 25 percent Bi_2Se_3. Typical p-type compounds are 75 percent Sb_2Te_3 and 25 percent Bi_2Te_3 with about 2 percent excess Te or Se. At a temperature difference of 40 °F, $(COP_c)_e$ ranges between 0.3 and 1.4. Cascading with up to three stages helps somewhat, but $(COP_c)_e$ remains low compared to the vapor cycle.

Thermoelectric systems are also quite expensive (Wood 1982). The design of the heat sink is a critical aspect of a good thermoelectric system. The heat sink must be able to reject heat load from the system. If the heat load is not rejected from the system, the temperature of the entire system will rise and the load temperature will increase. Increasing the current to maintain a specific load temperature results in a reduced efficiency. DC power supplies must also be included in initial costs. Most thermoelectric companies will custom design systems depending on the desired application.

RCA demonstrated a futuristic kitchen with a refrigerator based on thermoelectric cooling in the 1960s, but the system was never marketed. In the 1970s, some of the desktop computer monitors (linked to mainframes) had thermoelectric cooling. A medicine storage using a photovoltaic generator to drive a thermoelectric cooler (TEC) to pump 12 watts from a cooler at 0 °C to 40 °C at the cooling fin with a COP of 0.12 was developed. The price was estimated to be \$500 for the TEC and \$500 for the structure (Field 1979). Another model with 71 TE pellets in single-stage, with input of 33.64 W, at 7.41 V, had a COP of 0.3 at a temperature difference of 90 °C and heat removal of 10 W (Buist 1979). Space stations being planned by NASA will use vapor cycle refrigeration for cooling instead of TEC. Table 2 lists some typical applications of thermoelectric technology.

Table 2. Applications of thermoelectric technology.

Medical/ Laboratory	Industrial/ Commercial	Consumer	Military/ Aerospace
Temperature controlled therapy pads	Dewpoint hygrometers	Picnic box coolers and heaters	Electronic equipment cooling
Protein coolers	Osmometers	Air conditioned motorcycle helmets	Photo scanning equipment cooling
DNA amplifiers	Electronic enclosure coolers	Small refrigerators	Military avionics
Blood analyzers	Dehumidifiers		Infrared detectors
Constant temperature baths	CCD housing and cameras		Thermal viewers
Cold chambers	Integrated circuit coolers		Black body references
Cold-hot plates/ centrifuges	Temperature calibration systems		Space telescope cameras
Wafer probing stations	Laser diodes		Lightwave transmitters
	Graphic films		

Other systems. Many different manufacturers are developing alternative refrigeration systems. A fresh-air liquid desiccant absorption air-conditioning system is currently being developed. The units control temperature and humidity by use of a lithium bromide/water wash of the air. Units are expected to be in the 5- to 15-ton range rooftop market. There has also been discussion of the development of a Stirling free-piston heating and cooling system. Such a machine would use a Stirling cycle engine to drive a variable stroke linear compressor. Development on this system is currently stalled due to lack of funding. In reality, a Stirling cycle engine has external combustion and regenerator, and has the advantage of high efficiency (32+ percent) and low noise (Corey and Meacher 1985) and is worth pursuing. The free-piston drive is, perhaps, an unnecessary novelty.

Fuel cells (Hooie et al. 1992) might become important factors in the future energy scenario because of their high efficiencies. However, their exhaust gas temperatures (400 °F for phosphoric acid fuel cells to 1,800 °F for solid oxide fuel cells) call for cogeneration. An absorption refrigeration system will then be a compatible load of PAFC, for instance.

2.5 Geothermal Heat Sources and Sinks

The use of alternative heat sources (for the evaporator of a heat pump) and sinks (for the condenser of a chiller) has received increased attention in commercial and residential applications. In particular, ground and water source heat pump systems are currently being studied and improved. Many system configurations have been tried and system terminology often varies.

Using the ground or bodies of water as heat sources or sinks has several advantages. Even during extreme periods of ambient temperature, the ground remains at a relatively moderate temperature several feet below the surface. Average ground temperatures in the United States range from about 45 °F in the northern States to 75 °F in southern Florida (Pietsch 1990). Rizzuto (1994) reports that, in New York, the temperature of the earth 5 ft below the ground remains at a constant 45 to 50 °F over the entire year. The relatively constant temperature can be used as an efficient heat source for a heat pump and a heat sink for cooling. Surface water is capable of providing similar thermodynamic properties. Thermal stratification of water results in cold water remaining undisturbed at the bottom of deep lakes. In some instances this water is cool enough to provide cooling to buildings by simply being circulated through heat exchangers in the buildings, nullifying the need of a heat pump for cooling.

However, entering water temperatures must be kept lower than 55 °F to dehumidify the air effectively (Kavanaugh and Pezent 1990). Kavanaugh and Pezent (1990) also used data from several lakes in Alabama to evaluate the use of lakewater as a viable source/sink. They found that significant thermal stratification occurs in lakes deeper than 30 ft. In winter months, the lake temperatures remained constant between 45 and 55 °F at all depths. In summer months, water temperatures range between 80 and 90 °F at shallow depths. However, below the thermocline, water temperatures ranged between 45 and 55 °F. Control systems for water-to-air heat pumps are less complicated than air-to-air heat pumps because of the lack of a defrost cycle and stable water source temperatures (Kavanaugh 1991).

Hughes (1990) identifies various water source heat pump (WSHP) system configurations. Open-loop flow refers to systems for which liquid from the natural environment, such as from a pond or lake, is circulated through a water-to-air heat pump and is discharged directly back into the environment. Closed-loop flow refers to systems for which liquid is contained in a closed system and is continuously recirculated through a water-to-air heat pump.

WLHP. A water-loop heat pump (WLHP) system does not necessarily use a geothermal heat source or sink. A closed loop recirculates water through heat pumps after being heated (Howell and Zaidi 1990; Kuch 1990; Cooper 1990). The temperature of the loop is maintained between 60 and 90 °F. If the temperature of the loop approaches 60 °F, heat is added to the loop, usually by means of an electric or fossil fuel boiler. If the temperature of the loop approaches 90 °F heat is rejected from the loop, usually by means of an evaporative cooler or cooling tower. Water-source heat pumps use in WLHP systems are rated in the ARI Standard 320-86 at 85 °F for cooling performance ratings and 70 °F for heating performance ratings. Table 3 lists the COP of a WLHP system as compared to the ARI Standard rating conditions for the extreme values of water loop temperature (Pietsch 1990).

ARI Standard 325-85 is used to rate groundwater-source heat pumps and 330-90 is used to rate ground-source closed-loop heat pumps. These standards are not appropriate in some climates and need to be revised to reflect the efficiency of these systems more accurately (Kavanaugh et al. 1991). Lenarduzzi and Young (1990) report that a ground-source heat pump installation standard was adopted for Canada in 1989 that covers both open systems and closed-loop systems; a limit is the formation of subterranean ice lenses in certain soils.

2.5.1 *CLGC*

A closed-loop ground-coupled (CLGC) system is used to circulate liquid continuously through the heat pump and below the ground. In colder climates, an antifreeze solution is required as the circulating medium. Polyethylene or polybutylene pipe is configured in either vertical or horizontal loops. WLHPs are not adequate to operate in GCHP (ground coupled heat pump) systems because they are designed to operate in the narrow range of 60 to 90 °F. Heat pumps used in GCHP systems must be extended range water-to-air heat pumps (Kavanaugh 1992). Advantages of GCHPs are improved part-load efficiency and reduced parasitic power requirements because water has a higher thermal capacity than air and requires less pumping energy to be circulated (Kavanaugh et al. 1994). Further improvements in GCHP system design include the use of high efficiency reciprocating compressors and variable speed heat pumps.

2.5.2 CLWS

A closed-loop water-source (CLWS) system consists of a closed loop that is used to circulate water or antifreeze through a heat pump. Either groundwater or surface water is used as the heat source/sink. The loop is conditioned via a plate-and-fin heat exchanger. Disadvantages of CLWS systems are the risk of fouling on the outside coils located in bodies of water and the possibility that the coils may become damaged.

Table 3. comparison of COP to standard rated COP at various entering water temperatures.

Entering water temperature	Cooling efficiency compared to ARI rating condition	Heating efficiency compared to ARI rating condition
60°F	30% above	6% below
90°F	6% below	8% above

2.5.3 OLGW

An open-loop groundwater (OLGW) system allows groundwater from a well to be circulated through the pump. The water is then rejected to the surface for use as wash water. An alternative is an open-loop surface water system. Fouling of the heat exchangers could be a concern with open-loop systems. However, effective methods exist dealing with fouling, and only severe fouling will significantly degrade the performance (Kavanaugh and Pezent 1990).

2.5.4 CLGCHP

A case for consideration is the closed loop system using deep wells (Aldridge 1995) (alternately referred to as CLGCHP). Such a system was proposed for 4,003 units for family housing at Fort Polk, LA and was estimated to save the government $10 million over 20 years. A serious environmental concern is the water flow rate of 2.5 gpm/ton. Approximately 9,000 gpm of water will be reinjected into the aquifer after passing through the heat exchangers. Its effect on the water quality of the aquifer should be seriously considered.

2.5.5 GSHP

A direct expansion ground-source heat pump is under investigation. This system uses refrigerant piping buried in the ground to extract heat. A direct-expansion system eliminates the need for a secondary refrigerant loop and circulating pump, thus conserving heat transfer surface area and space occupied by the ground coil. Lenarduzzi and Bennet (1991) described such a system, which produced about 8 kW (27,600 Btu/h) of heat at a COP of 2.85. The system requires 28 lb of R-22. Spiral refrigerant coils are buried in three separate trenches, each measuring 6 ft deep and 55 ft long. Backfill material was used to prevent mechanical damage to the coils and to improve the thermal conductivity of the soil. A disadvantage is that the large refrigerant charge of a direct-expansion system is at risk of being lost to the environment in the event of damage to the ground coupled coils.

Installation costs of these systems are perhaps the most significant barriers for these new technologies. The first cost of a GCHP system includes 50 percent for the drilling, excavation, and piping. Compared to typical residential electric cooling/natural gas heating, $500 to $800 per ton has to be added for a horizontal system and $600 to $1,000 per ton for a vertical system with payback in 5 to 8 years. Vertical ground coils range from $2.50 to $7.00 per foot of bore. Bore lengths range between 125 ft per ton for cold climates to 300 ft per ton for warm climates. Pipe costs can range from $0.20 to $1.00 per foot of bore. Drilling costs can range from $1.00 to $12.00 per foot. A cost analysis given by Kavanaugh (1992) showed that items of piping, drilling, fittings, and grout gave per-ton costs of $464 (3/4-in. pipe) to $501 (1.5-in. pipe) for $1.5/ft drilling cost and $964 (3/4-in. pipe) to $926 (1.5-in. pipe) for $4/ft drilling, all on a per-ton basis.

Maintenance costs of WLHP and CLGC systems are reported to be inexpensive. For GCHPs to expand in the commercial market, it will be necessary for the HVAC industry and drilling/trenching industry to work together to make the installation of these systems competitive (Kavanaugh 1992).

The type of applications of GCHP is seen in a typical listing of systems installed and operated by a Pennsylvania firm (Kavanaugh 1992). Note that, of a total capacity of 3,045 tons installed with a total of 1,173 heat pumps and 708 bores, the average unit capacity was less than 6 tons and (the maximum) rarely more than 25 tons. Even in a large installation of 1,100 tons at a life care community, 527 heat pump units were installed with a total of 263 bores. The Fort Polk system completed in 1996 with geothermal source is the largest ever envisioned. The advantage is in the flexibility of small individual units that can be turned on or off (the same for a motel) at locations where heating or cooling is needed.

3 Baseline Case—Electric Motor-Driven Vapor-Compression Systems

The electric motor-driven, vapor-compression system was chosen as the baseline case since these systems dominate the cooling market and cover the widest range of ton sizes. In 1989, over 97 percent of the large (over 100-ton) liquid chillers shipped were electric motor-driven, vapor-compression systems (EPRI 1992). These systems use the vapor-compression cycle to achieve the refrigeration effect. The electric motor-driven systems typically have an advantage of relatively low first cost. Due to their popularity and predominating market share, it is also easier to find a variety of suppliers with a wide range of sizes from which to choose. Table 4 shows the manufacturers' data that was collected for these types of systems. The table shows that one of the main characteristics of this system is the type of compressor. The systems examined use centrifugal, reciprocating, screw, scroll, and rotary type compressors. Most of the cooling systems in large buildings are centrifugal or screw type machines (Beyene 1995). The type of compressor used can have an effect on system performance and type of refrigerant that can be used efficiently.

The table is divided into air-cooled and water-cooled condenser type units as this was found to be a factor in the first cost of the system. The first costs listed are for the base system only and do not include installation or ancillary equipment such as cooling towers and their associated piping. The higher tonnage systems typically use water as the heat rejection medium since it results in a much smaller, cheaper heat exchanger unit for the condenser than an air-cooled unit. Collected manufacturer's data shows no air-cooled units over 450 rated tons.

Table 4. Manufacturers' data on electric motor driven vapor-compression systems.

Mfr. Code	Refrigerant	Compressor	First Cost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
B	R-134A	Centrifugal		800 - 2300		40	85	6.1	Water
B	R-134A	Centrifugal		3000 - 7000		40	85	5.5	Water
B	R-134A	Centrifugal		200		40	85	5.6	Water
B	R-22	Centrifugal		300		40	85	5.6	Water
B	R-22	Centrifugal		350		40	85	5.6	Water
B	R-22	Centrifugal		400		40	85	5.6	Water
B	R-22	Centrifugal		450		40	85	5.6	Water
B	R-22	Centrifugal		500		40	85	5.6	Water
B	R-22	Centrifugal		550		40	85	5.6	Water
B	R-22	Centrifugal		600		40	85	5.6	Water

Mfr. Code	Refrigerant	Compressor	First Cost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
B	R-134A	Centrifugal		300		40	85	6.1	Water
B	R-134A	Centrifugal		350		40	85	6.1	Water
B	R-134A	Centrifugal		400		40	85	6.1	Water
B	R-134A	Centrifugal		450		40	85	6.1	Water
B	R-134A	Centrifugal		500		40	85	6.1	Water
B	R-134A	Centrifugal		550		40	85	6.1	Water
B	R-22	Screw		150		40	85	5.4	Water
B	R-22	Screw		185		40	85	5.4	Water
B	R-22	Screw		200		40	85	5.4	Water
B	R-22	Screw		230		40	85	5.4	Water
B	R-22	Screw		250		40	85	5.4	Water
B	R-22	Screw		275		40	85	5.4	Water
B	R-22	Screw		300		40	85	5.4	Water
B	R-22	Screw		325		40	85	5.4	Water
B	R-22	Screw		350		40	85	5.4	Water
B	R-22	Reciprocating		40		44	95	3.3	Air
B	R-22	Reciprocating		50		44	95	3.2	Air
B	R-22	Reciprocating		60		44	95	3.1	Air
B	R-22	Reciprocating		70		44	95	3.1	Air
B	R-22	Reciprocating		80		44	95	3.1	Air
B	R-22	Reciprocating		90		44	95	3.1	Air
B	R-22	Reciprocating		100		44	95	3.2	Air
B	R-22	Reciprocating		110		44	95	3.1	Air
B	R-22	Reciprocating		150		44	95	3.1	Air
B	R-22	Reciprocating		190		44	95	3.0	Air
B	R-22	Reciprocating		210		44	95	3.0	Air
B	R-22	Reciprocating		15		44	85	3.9	Air
B	R-22	Reciprocating		20		48	85	3.6	Air
B	R-22	Reciprocating		25		43	85	3.5	Air
B	R-22	Reciprocating		30		47	85	3.6	Air
B	R-22	Reciprocating		35		41	85	3.3	Air
B	R-22	Reciprocating		250		44	95	3.1	Air
B	R-22	Reciprocating		280		44	95	3.0	Air
B	R-22	Reciprocating		40		44	85	4.2	Water
B	R-22	Reciprocating		50		44	85	4.1	Water
B	R-22	Reciprocating		60		44	85	3.9	Water
B	R-22	Reciprocating		70		44	85	4.1	Water
B	R-22	Reciprocating		80		44	85	4.0	Water
B	R-22	Reciprocating		90		44	85	3.9	Water
B	R-22	Reciprocating		100		44	85	4.0	Water
B	R-22	Reciprocating		110		44	85	3.9	Water
B	R-22	Reciprocating		120		44	85	3.9	Water
B	R-22	Reciprocating		140		44	85	3.8	Water
B	R-22	Reciprocating		160		44	85	3.7	Water
B	R-22	Reciprocating		195		44	85	3.5	Water
B	R-22	Reciprocating		225		44	85	3.5	Water
B	R-22	Reciprocating		18		44	85	4.2	Water
B	R-22	Reciprocating		25		44	85	4.3	Water

Mfr. Code	Refrigerant	Compressor	First Cost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
B	R-22	Reciprocating		28		44	85	4.1	Water
B	R-22	Reciprocating		35		44	85	3.9	Water
B	R-22	Reciprocating		40		44	85	3.8	Water
C	R-22	Screw	$18,000	50	$360			4.4	Water
C	R-22	Screw	$29,000	100	$290			4.4	Water
C	R-22	Screw	$40,000	150	$267			4.4	Water
C	R-22	Screw	$62,000	225	$276			4.4	Water
C	R-22	Screw	$75,000	300	$250			4.4	Water
C	R-22	Screw	$27,000	100	$270				Water
C	R-22	Screw	$32,000	150	$213				Water
C	R-22	Screw	$36,000	180	$200				Water
C	R-22	Screw	$53,000	240	$221				Water
C	R-22	Screw	$65,000	330	$197				Water
C	R-22	Screw	$68,000	360	$189				Water
C	R-22	Screw	$92,000	480	$192				Water
C	R-22	Screw	$182,000	1100	$165				Water
C	R-22	Screw	$110,000	500	$220				Water
C	R-22	Screw	$120,000	700	$171				Water
C	R-22	Screw	$146,000	900	$162				Water
C	R-22	Reciprocating	$7,400	15	$493				Water
C	R-22	Reciprocating	$8,000	20	$400				Water
C	R-22	Reciprocating	$8,800	25	$352				Water
C	R-22	Reciprocating	$10,300	30	$343				Water
C	R-22	Reciprocating	$20,800	75	$277				Water
C	R-22	Reciprocating	$29,500	105	$281				Water
C	R-22	Reciprocating	$37,100	130	$285				Water
C	R-22	Reciprocating	$4,950	2	$2,475				Water
C	R-22	Reciprocating	$5,230	3	$1,743				Water
C	R-22	Reciprocating	$5,520	5	$1,104				Water
C	R-22	Reciprocating	$5,750	8	$719				Water
C	R-22	Reciprocating	$7,050	10	$705				Water
C	R-22	Reciprocating	$8,200	15	$547	44	95	2.7	Air
C	R-22	Reciprocating	$9,300	20	$465	44	95	2.6	Air
C	R-22	Reciprocating	$9,775	25	$391	44	95	2.7	Air
C	R-22	Reciprocating	$12,300	30	$410	44	95	2.7	Air
C	R-22	Reciprocating	$33,200	80	$415	44	95	2.6	Air
C	R-22	Reciprocating	$38,800	100	$388	44	95	2.6	Air
C	R-22	Reciprocating	$50,500	135	$374	44	95	2.7	Air
C	R-22	Reciprocating	$5,300	2	$2,650	44	95	1.9	Air
C	R-22	Reciprocating	$5,310	3	$1,770	44	95	2.9	Air
C	R-22	Reciprocating	$6,120	5	$1,224	44	95	2.7	Air
C	R-22	Reciprocating	$7,820	8	$978	44	95	3.0	Air
C	R-22	Reciprocating	$6,940	15	$463	45	95	3.3	Air
C	R-22	Reciprocating	$7,500	20	$375	45	95	3.1	Air
C	R-22	Reciprocating	$8,500	25	$340	45	95	3.0	Air
C	R-22	Reciprocating	$10,600	30	$353	45	95	3.1	Air
C	R-22	Reciprocating	$27,000	80	$338	45	95	3.1	Air
C	R-22	Reciprocating	$31,200	100	$312	45	95	3.1	Air

Mfr. Code	Refrigerant	Compressor	First Cost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
C	R-22	Reciprocating	$39,700	135	$294	45	95	3.1	Air
C	R-22	Screw	$21,300	40	$533	44	95	2.7	Air
C	R-22	Screw	$39,100	80	$489	44	95	2.7	Air
C	R-22	Screw	$48,300	120	$403	44	95	2.7	Air
C	R-22	Screw	$84,000	210	$400	44	95	2.8	Air
C	R-22	Screw	$91,400	270	$339	44	95	2.7	Air
C	R-22	Screw	$125,600	370	$339	44	95	2.6	Air
C	R-22	Screw	$145,000	420	$345	44	95	2.7	Air
C	R-22	Screw	$27,000	50	$540	45	95	3.0	Air
C	R-22	Screw	$44,000	100	$440	45	95	3.0	Air
C	R-22	Screw	$57,300	150	$382	45	95	2.8	Air
C	R-22	Screw	$30,400	45	$676				Air
C	R-22	Screw	$31,800	55	$578				Air
C	R-22	Screw	$58,300	105	$555				Air
C	R-22	Screw	$86,600	200	$433				Air
C	R-22	Screw	$109,000	260	$419				Air
P	R-22	Scroll	$15,141	20	$757	44	85	3.5	Air
P	R-22	Scroll	$16,985	25	$679	44	85	3.5	Air
P	R-22	Scroll	$19,028	30	$634	45	85	3.6	Air
P	R-22	Scroll	$26,702	40	$668	44	85	3.5	Air
P	R-22	Scroll	$29,456	50	$589	45	85	3.4	Air
P	R-22	Scroll	$34,169	60	$569	46	85	3.6	Air
P	R-22	Scroll		10		49	85	3.3	Air
P	R-22	Scroll		15		45	85	3.0	Air
P	R-22	Scroll	$15,141	20	$757	50	85	3.4	Air
P	R-22	Scroll	$15,002	25	$600	46	85	3.2	Air
P	R-22	Scroll	$17,398	30	$580	47	85	3.3	Air
P	R-22	Scroll	$19,999	40	$500	51	85	3.4	Air
P	R-22	Scroll	$22,665	50	$453	50	85	3.4	Air
P	R-22	Scroll	$26,619	60	$444	49	95	2.8	Air
P	R-22	Scroll	$13,700	20	$685	42	85	4.2	Water
P	R-22	Scroll	$14,142	25	$566	44	85	4.0	Water
P	R-22	Scroll	$16,413	30	$547	45	85	4.1	Water
P	R-22	Scroll	$22,255	40	$556	45	95	3.8	Water
P	R-22	Scroll	$22,559	50	$451	45	85	4.1	Water
P	R-22	Scroll	$27,693	60	$462	45	85	4.0	Water
P	R-11/R-123	Centrifugal	$94,176	200	$471	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$121,306	300	$404	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$125,576	400	$314	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$150,385	500	$301	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$175,815	600	$293	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$184,507	700	$264	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$199,187	800	$249	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$206,832	900	$230	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$242,696	1000	$243	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$263,214	1100	$239	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$285,021	1200	$238	38	100	6.4	Water
P	R-11/R-123	Centrifugal	$305,076	1300	$235	38	100	6.4	Water

Mfr. Code	Refrigerant	Compressor	First Cost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
P	R-11/R-123	Centrifugal	$305,079	1400	$218	38	100	6.4	Water
P	R-22	Screw	$31,053	70	$444	46	95	3.0	Air
P	R-22	Screw	$31,404	80	$393	45	95	3.0	Air
P	R-22	Screw	$34,908	90	$388	45	95	2.9	Air
P	R-22	Screw	$36,356	100	$364	45	95	2.8	Air
P	R-22	Screw	$38,990	110	$354	46	95	2.9	Air
P	R-22	Screw	$41,898	125	$335	47	95	2.9	Air
P	R-22	Screw	$59,518	130	$458	44	95	2.8	Air
P	R-22	Screw	$65,062	140	$465	44	95	2.8	Air
P	R-22	Screw	$68,226	155	$440	46	95	2.8	Air
P	R-22	Screw	$70,362	170	$414	46	95	2.7	Air
P	R-22	Screw	$73,923	185	$400	48	95	2.8	Air
P	R-22	Screw	$78,849	200	$394	48	95	2.7	Air
P	R-22	Screw	$85,267	215	$397	50	95	2.8	Air
P	R-22	Screw	$92,540	240	$386	47	95	2.8	Air
P	R-22	Screw	$101,945	270	$378	48	95	2.8	Air
P	R-22	Screw	$111,672	300	$372	49	95	2.7	Air
P	R-22	Screw	$128,375	340	$378	47	95	2.8	Air
P	R-22	Screw	$139,209	370	$376	48	95	2.8	Air
P	R-22	Screw	$150,496	400	$376	48	95	2.7	Air
P	R-22	Screw	$48,257	130	$371	45	85	5.2	Water
P	R-22	Screw	$52,740	150	$352	45	85	5.4	Water
P	R-22	Screw	$57,742	180	$321	45	85	5.3	Water
P	R-22	Screw	$61,596	215	$286	45	85	5.5	Water
P	R-22	Screw	$68,049	255	$267	45	85	5.6	Water
P	R-22	Screw	$72,173	300	$241	45	85	5.8	Water
P	R-22	Screw	$79,357	360	$220	45	85	5.5	Water
P	R-22	Screw	$90,139	450	$200	45	85	5.4	Water
P	R-22	Screw	$29,592	70	$423	42	85	4.1	Water
P	R-22	Screw	$31,022	80	$388	42	85	4.2	Water
P	R-22	Screw	$34,018	90	$378	44	95	3.7	Water
P	R-22	Screw	$37,748	100	$377	42	95	3.6	Water
P	R-22	Screw	$40,008	110	$364	42	95	3.6	Water
P	R-22	Screw	$42,988	125	$344	40	95	3.5	Water
R	R-11	Centrifugal	$72,300	150	$482	49	85	5.4	Water
R	R-11	Centrifugal	$80,200	200	$401	49	85	5.6	Water
R	R-11	Centrifugal	$98,400	300	$328	49	85	5.4	Water
R	R-11	Centrifugal	$102,800	400	$257	49	85	5.6	Water
R	R-11	Centrifugal	$145,800	600	$243	49	85	5.8	Water
R	R-11	Centrifugal	$178,400	800	$223	49	85	5.5	Water
R	R-22	Centrifugal	$203,000	1000	$203	49	85	5.6	Water
R	R-22	Centrifugal	$235,200	1200	$196	49	85	5.6	Water
R	R-22	Centrifugal	$292,500	1500	$195	49	85	5.6	Water
R	R-22	Centrifugal	$336,000	1600	$210	49	85	5.6	Water
R	R-22	Centrifugal	$352,800	1800	$196	49	85	5.6	Water
R	R-22	Centrifugal	$382,000	2000	$191	49	85	5.6	Water
R	R-22	Centrifugal	$361,200	2100	$172	49	85	5.6	Water
R	R-22	Reciprocating	$25,000	50	$500				Air

Mfr. Code	Refrigerant	Compressor	First Cost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
R	R-22	Reciprocating	$22,020	60	$367				Water
R	R-22	Reciprocating	$29,500	100	$295				Water
R	R-22	Reciprocating	$37,500	100	$375				Air
R	R-22	Reciprocating	$40,950	150	$273				Water
R	R-22	Reciprocating	$55,800	150	$372				Air
R	R-22	Reciprocating	$48,600	200	$243				Water
R	R-22	Reciprocating	$69,800	200	$349				Air
R	R-22	Reciprocating	$57,000	250	$228				Water
R	R-22	Reciprocating	$101,500	250	$406				Air
R	R-22	Reciprocating	$116,100	300	$387				Air
R	R-22	Reciprocating	$132,300	350	$378				Air
R	R-22	Reciprocating	$153,200	400	$383				Air
R	R-22	Reciprocating	$175,050	450	$389				Air
S	R22	Rotary	$559	1	$559			2.8	Air
T	R22	Rotary	$519	1	$519			2.6	Air

3.1 Refrigerants

The type of refrigerant used in the cycle can have a noticeable impact on the system COP. This is because, at the same conditions (pressure, temperature, quality, etc.), different refrigerants will have different thermodynamic properties, such as enthalpy. Table 5 compares the COP of a vapor-compression cycle refrigeration system using different refrigerants. The values were obtained assuming a T_e = 40 °F and a T_c = 90 °F with an overall compressor efficiency of 75 percent, which also took into account superheat, line pressure drops, and other losses.

The importance of this COP variation becomes more pronounced when one considers the elimination of CFC-based refrigerants due to their link to global warming and ozone depletion (a consideration that leads to the possible reintroduction of ammonia). Virtually all of the vapor-compression systems reviewed in this study use R-22 or R-134a. CFC production has stopped since 1996 and HCFC refrigerants, while somewhat better than CFCs from a global warming and ozone depletion standpoint, are at best an interim refrigerant. By international treaty, HCFC will be subject to a production freeze in 2015 and a total production ban by 2030 (EPRI 1992).

Table 5. Variation of COP with refrigerant type.

	100% η_c	75% η_c
Refrigerant	**$(COP_c)_e$**	**$(COP_c)_e$**
CFC-11	9.234	6.925
HCFC-22	8.644	6.483
HFC-134a	8.634	6.476
Ammonia	8.881	6.661

Table 5 implies that, as systems move from R-11 to R-22 and R-134a, the COP will decrease unless the system design is modified to maximize the performance of the new refrigerant. Another factor to consider is that not all refrigerants can be directly substituted into an existing system; component performance, wear, and materials compatibility also need to be considered when looking at replacement refrigerants. Even in these cases, it may become increasingly difficult to obtain the same level of COP without using new component and system technologies. Ammonia offers a non-CFC alternative, but brings with it additional safety concerns and a lowered performance.

The type of refrigerant used also has a particular bearing on the type of compressor used. Reciprocating compressors are best adapted to low specific volume flow rate and high pressure, whereas centrifugal compressors are suited for low pressures and high volume flow rate.

3.2 Relation to Temperature Conditions

Another factor that affects the COP as well as the ton rating of the system is the evaporator and condenser temperature. The system matrices list the evaporator and condenser temperatures quoted by the manufacturers along with the ton rating and COP. Notes that these temperatures can vary between manufacturers. A method for normalizing the various COP values based on the evaporator and condenser temperature difference was developed. Figure 2 shows the curve for the COP correction factor of the vapor-compression cycle based on R-134a as a function of the difference between T_c and T_e.

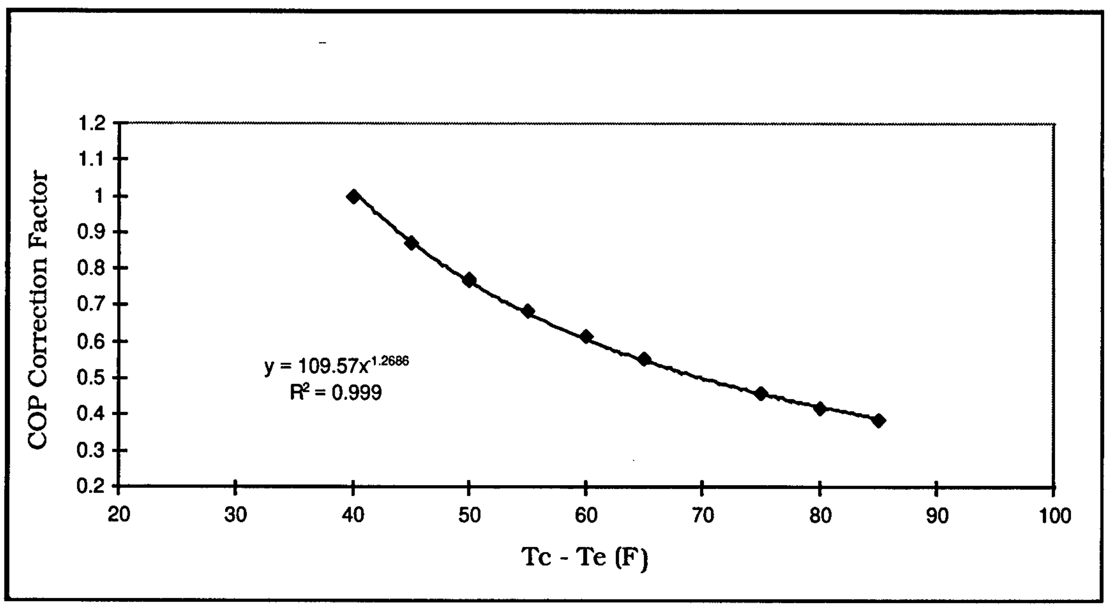

Figure 2. COP correction factor vs. condenser/evaporator temperature difference.

To determine the COP correction factor, the ideal COP was calculated for various condenser and evaporator combinations. The baseline temperatures chosen were T_e = 45 °F and T_c = 85 °F. The COP calculation at these temperatures was given the value of unity. The COP at other temperature differences were divided by the base COP value to determine the correction factor for that particular temperature difference. Normalizing the correction factor in this fashion allows its use even with systems having different refrigerants. If the COP of the systems under consideration are rated at other than a 40 °F temperature difference by dividing the stated COP by the correction factor given for the stated temperature difference, one can normalize the different system COPs at a common T_c and T_e temperature difference. For example, the system COP rated at a 60 °F temperature difference should be divided by approximately 0.61 to normalize the COP to a 40 °F temperature difference.

3.3 Compressors

The heart of any vapor-compression system is the compressor—the single-most expensive part of such a system. The four most common types of refrigeration compressors are the reciprocating, screw, centrifugal, and vaned. Each has its own advantages and disadvantages.

3.3.1 Reciprocating Compressors

Reciprocating compressors have positive displacement and consist of a piston moving in a cylinder with suction and discharge valves arranged such that pumping takes place. Reciprocating compressors range in size from less than 1 kW to 300 kW of refrigeration capacity and tend to have better efficiencies at part load conditions. A disadvantage of reciprocating compressors is their large physical size. Centrifugal and screw compressors are often preferred for high capacity systems due to their compactness.

3.3.2 Screw Compressors

Rotary screw compressors are also positive displacement machines. A female rotor drives a male rotor in a stationary housing. The refrigerant vapor enters one end of the compressor at the top and leaves the other end at the bottom. As the rotors rotate, trapped gas is moved circumferentially around the housing of the compressor. The volume in the cavity decreases and the gas is compressed. Peak efficiencies of screw compressors can be quite high. Screw compressors have a niche in the 300 kW to 500 kW capacity range and are efficient when operating near full load. Screw compressors have fewer moving parts and have a reputation for long life (Stoecker and Jones 1982).

3.3.3 Vane Compressors

Vane compressors are also positive displacement machines. Two basic types are roller (single-vane) type and the multiple vane type. Vane compressors are most often used in domestic applications including refrigerators, freezers, and air-conditioners. Compression is done by a rotor that rotates inside a cylinder.

3.3.4 Centrifugal Compressors

Centrifugal compressors are not positive displacement machines. Incoming refrigerant enters the eye of a spinning impeller and is thrown to the periphery of the impeller. The impeller blades impart a high velocity to the gas, which is converted to increased pressure in the diffuser. The pressure rise is determined by its tip speed. Centrifugals are most efficient for refrigeration systems with high capacities in the range of 200 to 10,000 tons of refrigeration. A two-stage centrifugal compressor facilitates the use of an economizer in the system.

4 Engine-Driven Vapor-Compression Systems

Gas engine-driven chillers and heat pumps use the same thermodynamic cycle as conventional electric motor-driven systems. The only difference is that the compressor is powered by a natural gas engine rather than an electric motor. Engine-driven systems have several potential advantages over electric motor-driven units. Among them are variable speed operation, high part load efficiency, high temperature waste heat recovery from the engine, and reduced annual operating costs (American Gas Cooling Center 1994). Table 6 shows the manufacturers' data compiled for gas engine-driven cooling systems. While the listing gives ratings up to 4000 tons, operating experience so far appears to be limited to 1000 tons or less.

Table 6. Manufacturers' data on gas engine driven cooling systems.

Mfr. Code	Refrigerant	Compressor	FirstCost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
A	R-22	Reciprocating		50				1.38	Water
A	R-22	Reciprocating		150				1.52	Water
A	R-22	Screw		250					Water
A	R-22	Reciprocating		300					Water
A	R-22	Screw		500					Water
A	R-22	Screw		750					Water
A	R-22	Screw		1040				1.32	Water
B	R-22	Reciprocating		25				1.0	Air
D	R-22	Reciprocating	$95,000	95	$1,000				Water
D	R-22	Reciprocating	$115,000	140	$821				Water
D	R-22	Reciprocating	$155,000	195	$795				Water
D	R-22	Screw	$190,000	250	$760			1.57	Water
D	R-22	Screw	$210,000	300	$700				Water
D	R-22	Reciprocating	$200,000	315	$635				Water
D	R-22	Screw	$365,000	400	$913			1.79	Water
D	R-22	Screw	$310,000	500	$620				Water
D	R-22	Screw	$370,000	650	$569				Water
D	R-22	Screw	$410,000	800	$513			1.81	Water
D	R-22	Screw	$475,000	900	$528				Water
D	R-22	Screw	$560,000	1000	$560				Water
D	R-22	Screw	$650,000	1100	$591				Water
D	R-22	Centrifugal		2000					Water
D	R-22	Centrifugal		4000					Water
D	R-22	Centrifugal		6000					Water
L	R-22	Reciprocating		50 - 300				1 -2	Water

Mfr. Code	Refrigerant	Compressor	FirstCost	Ton Rating	Cost/Ton	T_e(F)	T_c(F)	COP	Cooling
L	R-134A	Screw		50 - 4000				1 -2	Water
M	R-22	Reciprocating	$60,000	60	$1,000			0.95	Water
M	R-22	Reciprocating	$68,775	75	$917			1.3	Water
M	R-22	Screw	$79,420	110	$722			0.85	Water
M	R-22	Screw	$80,040	120	$667			0.82	Water
M	R-22	Screw	$79,875	125	$639			1.3	Water
M	R-22	Screw	$75,000	150	$500			1.2	Water
M	R-22	Screw	$80,000	160	$500			1.2	Water
M	R-22	Screw	$85,000	170	$500			1.2	Water
M	R-22	Screw	$170,000	340	$500			1.2	Water
M	R-22	Screw	$242,500	485	$500			1.2	Water
M	R-22	Screw	$350,000	700	$500			1.2	Water
N	R-22	Reciprocating	$21,925	15	$1,462			0.8	Air
N	R-22	Reciprocating	$27,200	25	$1,088			0.8	Air
R	R-22	Reciprocating	$7,000	3	$2,333			0.9	Air
R	R-22	Reciprocating	$8,000	4	$2,000			0.9	Air

4.1 Technical Comparison to Base Case

It has been determined (see Soo 1956) that a mechanical refrigeration system-driven by an engine or motor from an electric power generation of efficiency $\eta_e = (W/Q_s)$ has an equivalent COP based on a heat supply given by:

$$\left(COP_c\right)_s = \left(Q_c / Q_s\right) = \left(Q_c / W\right)\left(W / Q_s\right) = \left(COP_c\right)_e \eta_e \qquad \textbf{Eq 15}$$

in which η_e is based on the ambient temperature in the refrigeration mode. The coefficient of performance of a heat pump-driven by an engine using a heat source (gas or steam) is:

$$\left(COP_h\right)_s = \left(Q_h / Q_s\right) = \left(Q_h / W\right) / \left(W / Q_s\right) = \left(COP_h\right)_e \eta_e' = \left(COP_c\right)_s + \eta_e' \qquad \textbf{Eq 16}$$

in which η_e' is the engine efficiency based on the ambient temperature in the heating mode. As an example of a practical heat pump, let $(COP_h)_e$ be equal to 5 and the engine efficiency η_e' be equal to 30 percent. This would result in a $(COP_h)_s$ of 1.5. This implies that 1.5 Btu of heat could be made available for 1 Btu of fuel energy supplied. This is superior to absorption units operated as heat pumps and electric motor-driven heat pumps. Equivalent efficiency in the case of an electric drive is nearly 30 percent in terms of fuel conversion in a power plant of nearly 34 percent efficiency with transmission losses to the user and average motor efficiency. Gas engine-driven heat pumps have thermodynamic characteristics that give them an advantage over many traditional cooling technologies.

Much work has been done in Japan since 1979 towards the development of small gas engine-driven heat pumps. Gas utilities, engine manufacturers, and heat pump manufacturers have combined resources and have been supported by Japan's Ministry of International Trade and Industry (MITI) in developing this new technology. This development has been motivated by several advantages, some of which are also relevant in the United States.

Several potential advantages of these technologies over the base-case, motor-driven, vapor-compression cycle are readily identified. The first is to establish technologies using alternative energy sources such as natural gas to ensure a long-term energy supply. The second is energy conservation. Natural gas, loses only a small percentage of its energy in its transmission to the home. Losses could be high when fuel energy is first converted to electricity and transported to the user. According to Nowakowski et al. (1992), the overall source-to-end-use efficiency for engine-driven gas heat pumps is 130 percent compared to 85 percent for electric heat pumps assuming a 34 percent power plant efficiency. A third advantage of these technologies is that gas-fired cooling can shift energy demand in the summer from electricity to gas and ease the peak load for electricity. (Gas utilities have a surplus of energy during the summer.) A fourth advantage is possible reduction of air pollution. Combustion of natural gas produces the least amount of carbon dioxide of the fossil fuels when compared to that from power plants of utilities. The former produces about 45 percent less carbon dioxide than coal and 30 percent less carbon dioxide than oil for the same amount of energy input. Gas engine heat pumps also provide environmental advantages because sulfur dioxide emissions are low, a characteristic that helps to prevent acid rain (Nowakowski et al. 1992). A fifth advantage is reduced annual operating costs in areas that have high electric demand charges. No fewer than 12 field tests were conducted in North America since 1992, including the test at Fort Sam Houston, TX, under contract by Pacific Northwest Laboratory (1994). The one at WGNAS (1993) will be analyzed in detail in this report.

4.2 Developmental Activities

Fifteen Japanese companies established a consortium in 1981 for developing small gas engine heat pumps with the support of MITI. Three major Japanese utilities and three gas engine-driven heat pump manufacturers have been successfully marketing small commercial GHP's in Japan since 1987 (Miyairi 1989). Sizes being mass produced range from 1.3 to 16 tons (Yokoyama 1992). A natural gas engine-driven heat pump was also developed for light commercial applications in the United States by the Gas Research Institute, Chicago, and

Aisin Seiki Company Limited, Kariya, Japan (Nowakowski et al. 1992). Technical difficulties had to be overcome to produce machines that could be successfully marketed.

These difficulties represent potential disadvantages of gas engine cooling equipment over electric motor-driven units. Engine durability was identified as the primary obstacle. A machine life of at least 10 years would require a minimum durability for the engine of 20,000 hours of operation (Kazuta 1989). This level of durability is approximately equivalent to an automobile traveling 1.25 million miles (Miyairi 1989). Automobile, motorcycle, and diesel engines were modified and used as well as gas engines together with general purpose compressors (Yokoyama 1992). The engines provide the capability of variable engine speed and heat pump capacity to match the load. In capacities below 10 tons, gas engine-driven refrigeration equipment differs from electric motor-driven equipment in that gas engine systems have open type compressors like those in automobiles, whereas electric systems have hermetically sealed motor-compressor drives. All of the gas engine-driven systems use HCFC-22 as the refrigerant. Kaneko et al. (1992) cited that, for a recently developed 4-ton unit, the valve seats, piston rings, piston profile, valve stem seals, and engine oil characteristic and sump capacity, among others, had to be modified to achieve the desired engine durability. Table 7 lists typical specifications of a 5-ton cooling system given by Nowakowski et al. (1992).

Other objectives in the development of these systems included achievement of high engine thermal efficiency, reduction of engine noise (varying between 27 dB to 72 dB) and vibration, and reduction of required annual maintenance. Annual maintenance requirements are significantly higher than those of conventional electric motor-driven cooling equipment. Kazuta (1989) described the development of a 1.3-ton gas engine heat pump and cited the maintenance requirements (Table 8).

Engine-related problems were encountered frequently in Japan once the engine-driven heat pumps were operating in the field. Defective switch contacts and refrigerant pipe breakage due to engine vibration were also seen very often (Yokoyama 1992). Improvements from analysis of data of problems in the market and implementation of appropriate countermeasures in the designs have been made; recent models have had significantly less trouble than before (Yokoyama 1992). Gas engine-driven equipment requires more frequent maintenance by knowledgeable technicians than conventional electric motor-driven equipment.

Table 7. Specifications of a 5-ton gas engine driven heat pump.

Parameter	Specification
Cooling COP	0.9 at 95°F outdoor temperature
Heating COP	1.8 at 47°F outdoor temperature
Operating temperature range:	
Heating mode	-18°F to 80°F outdoor
Cooling mode	45°F to 115°F outdoor
Electric power	1.3 kW, AC 220 V, single phase
Fuel	Natural gas HHV 1,070 Btu/cu. ft. LHV 961 Btu/cu. ft.
Refrigerant	HCFC-22
Engine	Water-cooled 4-cycle, in-line, 3-cyl., 547 cc
Oil capacity	10 L
Engine speed	1,200 to 2,800 RPM
Starting	Starting motor, AC/DC converter
Compressor	60 cc/rev., scroll
Noise	72 dB

Table 8. Maintenance items of a small gas engine heat pump.

Maintenance Item	Interval
Exchange of engine oil	Every 4,000 hours
Additional supply of engine oil	Every 2,000 hours
Exchange of spark plug	Every 2,000 hours
Exchange of air filters	Every 2,000 hours
Exchange of oil filter	Every 8,000 hours
Exchange of V belt	Every 10,000 hours
Exchange of coolant	Every 8,000 hours

An advantage of engine-driven cooling equipment over electric motor-driven is the ability to use the heat rejected from the engine during operation. During heating operation, the recovered heat can be used to supplement the refrigeration cycle. The use of the engine waste heat results in greater operating efficiency compared to similar electric motor-driven units. When the heat emitted by the engine is used to prevent frosting, the $(COP_h)_s$ is reduced to value of 1. During cooling operation, the heat could be used to produce hot water (Nowakowski et al. 1992).

For chillers in the capacity range of 5 tons, on-off control is desirable, as in the case of electrical-driven chillers. However, frequent engine starts and stops have not been reported extensively in the literature. Yokoyama (1992) was the only one who suggested that the durability of the starter be tested by cranking it in excess of 100,000 times and that it is preferable to isolate the compressor load during starting by the use of a clutch. Units above 5 tons may call for unloader on the compressor valves with engine idling at half speed rather than stopping the engine when the demand is off. Kazuta (1989) mentioned the occurrence of large amplitude vibrations at engine starting and stopping and the use of an AC self-starting motor for a 1.3 RT unit. Two, 15-ton, natural gas engine-driven rooftop air-conditioners were recently installed and monitored at the Willow Grove Naval Air Station. Each unit had a three-stage starting sequence, which could increase engine speed from 1,200 RPM to 2,200 RPM. A 12-V DC starter was used (WGNAS 1993).

Significant disadvantages of gas engine-driven cooling equipment compared to the base case electric motor-driven units are the high initial cost and maintenance needs. Installed costs of gas engine-driven heat pumps in Japan quoted by Yokoyama (1992) in 1991 dollars ranged from $4,077/ton for a 1.3-ton unit to $2,593/ton for a 13-ton unit. Installed cost for the engine-driven air-conditioners at Willow Grove were $5,198/ton. A manufacturer quote of this equipment placed the uninstalled unit cost at $1,462/ton. Quotes from other U.S. manufacturers of uninstalled costs of gas engine-driven cooling systems range from around $500/ton for a 700-ton system to $2,333/ton for a 3-ton unit.

According to WGNAS (1993), gas powered units are usually cost-effective at a site with electric rates greater than 15 cents/kWh and gas rates less than 70 cents/therm (or $7/MMBtu). Gas engine-driven equipment has thermodynamic advantages over more conventional systems and potential added benefits such as reduced emissions. However, the many disadvantages include such considerations as high initial and maintenance costs. All factors should be considered when choosing equipment for a specific application. Applications to barracks, mess halls, and clubs appear practical—from the standpoint of efficiency. However, until something is done to correct the negative aspects of gas-driven equipment, it will difficult to win the acceptance of a scenario of rows of houses, each with an engine-driven outdoor chiller, running all night along with an exhaust muffler giving off exhaust gas. Each unit will have noise at least double that of a motor-driven outdoor chiller.

5 Absorption Systems

An absorption chiller system performs cooling using the same basic vapor-compression cycle except that the mechanical compressor of the vapor-compression system is replaced with components that provide compression by heating the solution. The vast majority of systems examined for air-conditioning use water as the refrigerant and lithium bromide ($LiBr-H_2O$) as the carrier or absorbent. With water as the refrigerant, the process takes place at much lower pressures than in a mechanical vapor-compression system at corresponding temperatures. Some chiller systems use water as the carrier and ammonia as the refrigerant. The cycle operation is basically the same for both types of units, but may operate at an evaporator temperature below 32 °F.

Figure 3 shows a schematic of a single-effect $LiBr-H_2O$ absorption chiller. High-pressure liquid water from the condenser is throttled down to the evaporator pressure (typically around 6mm Hg absolute to operate the evaporator at 40 °F). At this low pressure, the refrigerant liquid that is sprayed into the evaporator absorbs heat from the chilled water loop and evaporates. The evaporation process takes place at low temperatures due to the low absolute pressure maintained in the evaporator. This refrigerant vapor goes into the absorber where it is absorbed, in this case by LiBr. As the concentrated solution absorbs more refrigerant, its absorption ability decreases. The weak absorbent solution is pumped to the generator where heat is input to drive off the refrigerant. The ensuing strong solution returns to the absorber through a liquid-to-liquid heat exchanger that preheats the weak solution going to the generator. The hot refrigerant vapor from the generator is cooled to a liquid in the condenser. The liquid refrigerant is passed through the expansion valve and the cycle repeats. Single-effect systems typically operate at a $(COP_c)_s$ of around 0.7.

Double-effect absorption systems achieve improved efficiency by using two separate generators, which allows recovery and reuse of part of the heat input into the first-stage generator (Figure 4). The second generator drives off additional refrigerant and is itself driven by heat from the first-stage generator's refrigerant, which provides 30 to 40 percent more refrigerant than a single-effect system (American Gas Cooling Center 1994). Double-effect systems can be configured either in series or parallel flow. The difference between the two systems is the fluid path taken by the solution through the generators. The double-effect systems examined were rated at a $(COP_c)_s$ ranging from 0.95 to 1.4.

Table 9 shows the manufacturers' data collected for absorption systems. Except where indicated all systems are LiBr-H_2O type units. These types of units were first developed in the United States around 1945 and enjoyed success in the cooling market until around 1975. The market downturn was due to a number of factors, including low electric costs, rapid development of new vapor-compression technologies, and a brief "shortage" of natural gas in the 1970s (Wilkinson 1994). New absorption technologies, energy costs, and environmental concerns all contribute to absorption systems being viable candidates for building cooling.

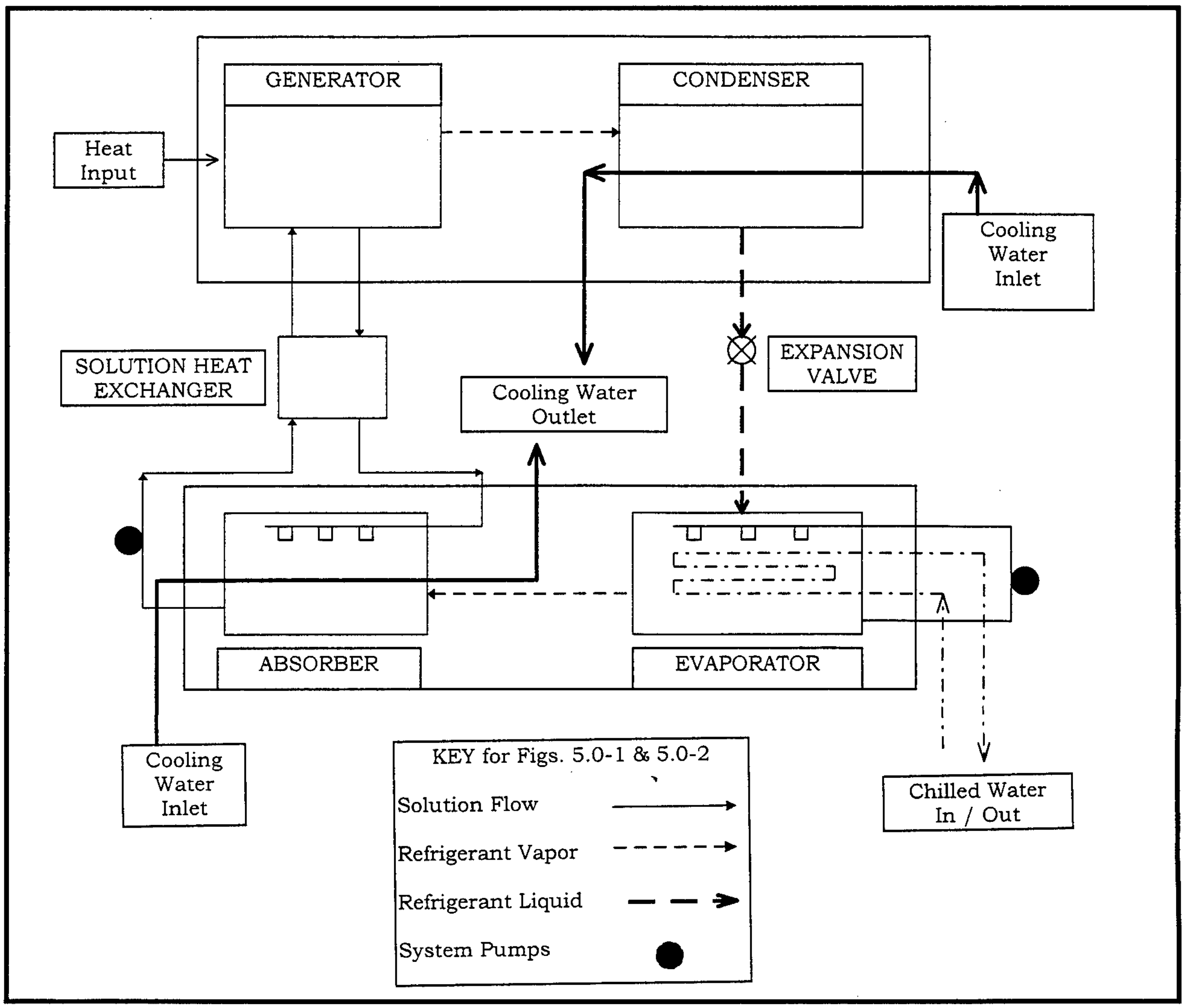

Figure 3. Single-effect absorption system.

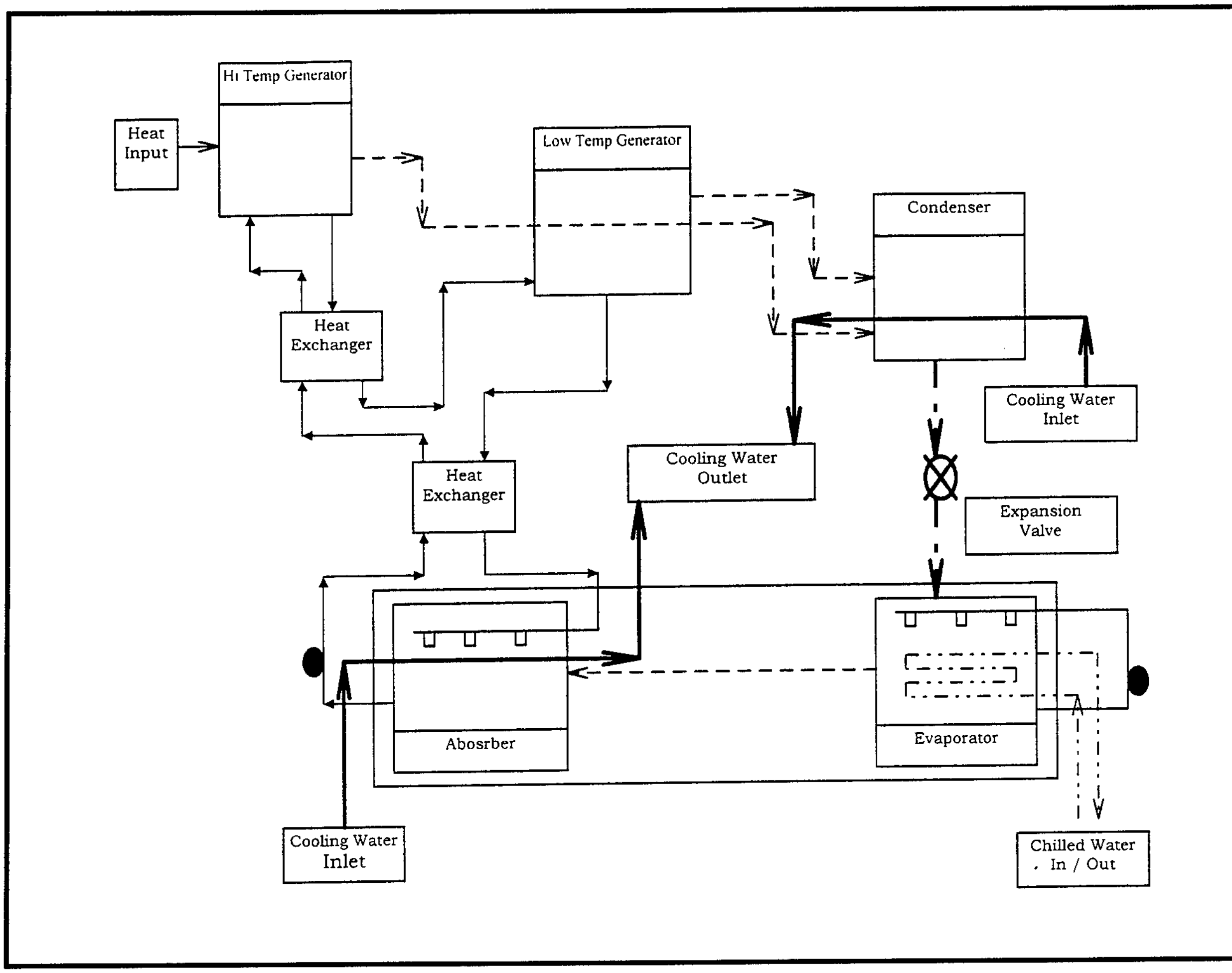

Figure 4. Double-effect absorption system (series flow).

Table 9. Manufacturers' data on absorption cooling systems.

Mfr. Code	Cycle	Solution	First Cost	Ton Rating	Cost/Tn	T_e(F)	T_c(F)	COP	Cooling	Heat
B	2 Effect	LiBr-Water	$126,000	135	$933	49	85	0.97	Water	Direct Fired
B	2 Effect	LiBr-Water	$207,500	500	$415	49	85	0.97	Water	Direct Fired
B	2 Effect	LiBr-Water	$552,500	800	$691	49	85	0.97	Water	Direct Fired
B	2 Effect	LiBr-Water	$678,900	1000	$679	49	85	0.97	Water	Direct Fired
B	1 Effect	LiBr-Water	$64,900	108	$601	49	85	0.7	Water	Steam
B	1 Effect	LiBr-Water	$102,550	326	$315	49	85	0.7	Water	Steam
B	1 Effect	LiBr-Water	$154,900	618	$251	49	85	0.7	Water	Steam
B	1 Effect	LiBr-Water	$160,700	680	$236	49	85	0.7	Water	Steam
B	2 Effect	LiBr-Water	$110,400	100	$1,104	49	85	1.2	Water	Steam
B	2 Effect	LiBr-Water	$135,200	145	$932	49	85	1.2	Water	Steam
B	2 Effect	LiBr-Water	$148,500	170	$874	49	85	1.2	Water	Steam
B	2 Effect	LiBr-Water	$242,700	500	$485	49	85	1.2	Water	Steam
B	2 Effect	LiBr-Water	$314,800	800	$394	49	85	1.2	Water	Steam
B	2 Effect	LiBr-Water	$371,100	1000	$371	49	85	1.2	Water	Steam
B	2 Effect	LiBr-Water	$548,000	1500	$365	49	85	1.2	Water	Steam
B	2 Effect	LiBr-Water	$620,700	1700	$365	49	85	1.2	Water	Steam
F	2 Effect	LiBr-Water	$56,000	80	$700	44	85	0.95	Water	Direct Fired

Mfr. Code	Cycle	Solution	First Cost	Ton Rating	Cost/Tn	T_e(F)	T_c(F)	COP	Cooling	Heat
F	2 Effect	LiBr-Water	$103,500	100	$1,035	49	90	1.0	Water	Direct Fired
F	2 Effect	LiBr-Water	$234,000	240	$975	49	90	1.0	Water	Direct Fired
F	2 Effect	LiBr-Water	$330,000	400	$825	49	90	1.0	Water	Direct Fired
F	2 Effect	LiBr-Water	$576,000	800	$720	49	90	1.0	Water	Direct Fired
F	2 Effect	LiBr-Water	$671,000	1100	$610	49	90	1.0	Water	Direct Fired
F	2 Effect	LiBr-Water	$750,000	1500	$500	49	90	1.0	Water	Direct Fired
F	2 Effect	LiBr-Water	$34,000	20	$1,700	44	85	0.95	Water	Direct Fired
F	2 Effect	LiBr-Water	$38,250	30	$1,275	44	85	0.95	Water	Direct Fired
F	2 Effect	LiBr-Water	$44,000	40	$1,100	44	85	0.95	Water	Direct Fired
F	2 Effect	LiBr-Water	$45,000	50	$900	44	85	0.95	Water	Direct Fired
F	2 Effect	LiBr-Water	$48,000	60	$800	44	85	0.95	Water	Direct Fired
F	2 Effect	LiBr-Water	$52,500	70	$750	44	85	0.95	Water	Direct Fired
F	2 Effect	LiBr-Water	$90,000	100	$900	49	90	1.4	Water	Steam
F	2 Effect	LiBr-Water	$204,000	240	$850	49	90	1.4	Water	Steam
F	2 Effect	LiBr-Water	$280,000	400	$700	49	90	1.4	Water	Steam
F	2 Effect	LiBr-Water	$480,000	800	$600	49	90	1.4	Water	Steam
F	2 Effect	LiBr-Water	$550,000	1100	$500	49	90	1.4	Water	Steam
F	2 Effect	LiBr-Water	$600,000	1500	$400	49	90	1.4	Water	Steam
H	1 Effect	Water-NH3		4		50	95	0.48	Air	Direct Fired
H	1 Effect	Water-NH3		5		50	95	0.48	Air	Direct Fired
H	1 Effect	Water-NH3		3		50	95	0.62	Air	Direct Fired
P	2 Effect	LiBr-Water	$132,000	100	$1,320			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$140,000	120	$1,167			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$153,000	150	$1,020			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$167,000	180	$928			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$178,000	200	$890			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$196,000	240	$817			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$239,000	300	$797			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$264,000	350	$754			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$298,000	400	$745			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$319,000	450	$709			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$355,000	500	$710			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$380,000	550	$691			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$436,000	600	$727			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$469,000	700	$670			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$527,000	800	$659			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$578,000	900	$642			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$623,000	1000	$623			1.0	Water	Direct Fired
P	2 Effect	LiBr-Water	$664,000	1100	$604			1.0	Water	Direct Fired
P	1 Effect	LiBr-Water	$89,000	112	$795			0.68	Water	Steam
P	1 Effect	LiBr-Water	$93,000	129	$721			0.68	Water	Steam
P	1 Effect	LiBr-Water	$96,000	148	$649			0.68	Water	Steam
P	1 Effect	LiBr-Water	$100,000	174	$575			0.68	Water	Steam
P	1 Effect	LiBr-Water	$104,000	200	$520			0.68	Water	Steam
P	1 Effect	LiBr-Water	$110,000	228	$482			0.68	Water	Steam
P	1 Effect	LiBr-Water	$118,000	256	$461			0.68	Water	Steam
P	1 Effect	LiBr-Water	$125,000	294	$425			0.68	Water	Steam
P	1 Effect	LiBr-Water	$134,000	354	$379			0.68	Water	Steam

Mfr. Code	Cycle	Solution	First Cost	Ton Rating	Cost/Tn	T_e(F)	T_c(F)	COP	Cooling	Heat
P	1 Effect	LiBr-Water	$135,000	385	$351			0.68	Water	Steam
P	1 Effect	LiBr-Water	$145,000	420	$345			0.68	Water	Steam
P	1 Effect	LiBr-Water	$147,000	465	$316			0.68	Water	Steam
P	1 Effect	LiBr-Water	$155,000	520	$298			0.68	Water	Steam
P	1 Effect	LiBr-Water	$168,000	590	$285			0.68	Water	Steam
P	1 Effect	LiBr-Water	$178,000	665	$268			0.68	Water	Steam
P	1 Effect	LiBr-Water	$211,000	750	$281			0.68	Water	Steam
P	1 Effect	LiBr-Water	$280,000	852	$329			0.68	Water	Steam
P	1 Effect	LiBr-Water	$303,000	955	$317			0.68	Water	Steam
P	1 Effect	LiBr-Water	$331,000	1125	$294			0.68	Water	Steam
P	1 Effect	LiBr-Water	$380,000	1250	$304			0.68	Water	Steam
P	1 Effect	LiBr-Water	$414,000	1465	$283			0.68	Water	Steam
P	1 Effect	LiBr-Water	$452,000	1660	$272			0.68	Water	Steam
P	2 Effect	LiBr-Water	$263,000	385	$683			1.2	Water	Steam
P	2 Effect	LiBr-Water	$286,000	465	$615			1.2	Water	Steam
P	2 Effect	LiBr-Water	$304,000	520	$585			1.2	Water	Steam
P	2 Effect	LiBr-Water	$325,000	590	$551			1.2	Water	Steam
P	2 Effect	LiBr-Water	$346,000	665	$520			1.2	Water	Steam
P	2 Effect	LiBr-Water	$393,000	750	$524			1.2	Water	Steam
P	2 Effect	LiBr-Water	$422,000	852	$495			1.2	Water	Steam
P	2 Effect	LiBr-Water	$456,000	955	$477			1.2	Water	Steam
P	2 Effect	LiBr-Water	$502,000	1125	$446			1.2	Water	Steam
Q	2 Effect	LiBr-Water	$28,600	30	$953	49	90	0.95	Water	Direct Fired
Q	2 Effect	LiBr-Water		40		49	90	0.95	Water	Direct Fired
Q	2 Effect	LiBr-Water	$40,800	50	$816	49	90	0.95	Water	Direct Fired
Q	2 Effect	LiBr-Water		60		49	90	1.0	Water	Direct Fired
Q	2 Effect	LiBr-Water		80		49	90	1.0	Water	Direct Fired
Q	2 Effect	LiBr-Water	$71,800	100	$718	49	90	1.0	Water	Direct Fired
Q	1 Effect	LiBr-Water	$23,200	5	$4,640	48	85	0.7	Water	Steam
Q	1 Effect	LiBr-Water	$23,200	7.5	$3,093	48	85	0.7	Water	Steam
Q	1 Effect	LiBr-Water	$23,200	10	$2,320	48	85	0.7	Water	Steam
R	1 Effect	LiBr-Water	$70,800	120	$590			0.69	Water	Steam
R	1 Effect	LiBr-Water	$74,245	155	$479			0.69	Water	Steam
R	1 Effect	LiBr-Water	$81,385	205	$397			0.69	Water	Steam
R	1 Effect	LiBr-Water	$118,490	410	$289			0.69	Water	Steam
R	1 Effect	LiBr-Water	$153,633	617	$249			0.69	Water	Steam
R	1 Effect	LiBr-Water	$196,912	794	$248			0.69	Water	Steam
R	1 Effect	LiBr-Water	$259,350	1235	$210			0.69	Water	Steam
R	1 Effect	LiBr-Water	$282,285	1377	$205			0.69	Water	Steam
R	2 Effect	LiBr-Water	$134,200	200	$671			0.92	Water	Direct Fired
R	2 Effect	LiBr-Water	$245,600	400	$614			0.92	Water	Direct Fired
R	2 Effect	LiBr-Water	$276,500	500	$553			0.92	Water	Direct Fired
R	2 Effect	LiBr-Water	$309,600	600	$516			0.92	Water	Direct Fired
R	2 Effect	LiBr-Water	$388,500	700	$555			0.92	Water	Direct Fired
R	2 Effect	LiBr-Water	$441,600	800	$552			0.92	Water	Direct Fired
R	2 Effect	LiBr-Water	$580,000	1000	$580			0.92	Water	Direct Fired
R	2 Effect	LiBr-Water	$216,920	440	$493			1.19	Water	Steam
R	2 Effect	LiBr-Water	$289,800	600	$483			1.19	Water	Steam

Mfr. Code	Cycle	Solution	First Cost	Ton Rating	Cost/Tn	T_e(F)	T_c(F)	COP	Cooling	Heat
R	2 Effect	LiBr-Water	$389,600	800	$487			1.19	Water	Steam
R	2 Effect	LiBr-Water	$486,000	1000	$486			1.19	Water	Steam
R	2 Effect	LiBr-Water	$624,000	1500	$416			1.16	Water	Steam

5.1 Technical Comparison to Base Case

The ideal $(COP_c)_s$ was presented as that obtained from a Carnot engine driving a Carnot refrigerator. However, Wang et al. (1992) commented that the ideal absorption cycle is not a pure combination of an independent heat engine and refrigerator since energy exchange between the heat engine and refrigerator takes place, not as mechanical work, but in the form of heat. The expression for ideal COP is Carnot with four temperature levels, which can be inferred to be:

$$(COP)_{ideal} = \left[\left(T_g - T_a\right)/T_g\right]\left[T_e/\left(T_c - T_e\right)\right] \qquad \textbf{Eq 17}$$

in which T_g is the generator temperature, T_c is the condenser temperature, T_e is the evaporator temperature and T_a is the absorber temperature. The above relations may provide an accurate view of the upper COP limit for an absorption system operating at the defined temperatures and provide additional avenues for the improvement of performance of the system. One can also see from this relation that the COP will increase with increasing generator temperature (or increasing evaporator temperature) and will decrease with increasing absorber temperature.

The $(COP_c)_s$ for an absorption system was defined in Section 2.0 as:

$$\left(COP_c\right)_s = Q_c / Q_s \qquad \textbf{Eq 18}$$

In the case of the absorption systems, Q_c is the effective cooling obtained from the system and Q_s is the heat input to the generator. This is analogous to the base case in which the value of the net work into the system compressor is generated from Q_s. As one examines the manufacturer data collected on the various systems, it is apparent that the electric motor-driven, vapor-compression systems (Table 4) provide a much higher COP than the absorption system. This could mistakenly cause someone to conclude that the absorption systems cannot compete with the base systems outlined in this study.

The above discussion on the ideal COP for the absorption systems leads one to recognize, as Wilkinson (1994) points out, that these systems combine the power cycle and refrigeration cycle into one unit. With the motor-driven vapor-compression systems, the power cycle is located at the electric utility company facility and the refrigeration cycle is on site. This arrangement helps to account for the lower initial cost of the base case systems as well as the wide disparity of cooling performance capability between the two systems. The vapor produced in the power cycle of the absorption system is used directly by the condenser in the refrigeration cycle. This interconnection between the power and refrigeration cycle forces the power cycle to operate within the fluid and pressure difference restrictions dictated by the refrigeration cycle.

The absorption systems do offer some attractive advantages over the base case systems. These include use of alternative fuels (natural gas) and heat sources (waste heat from industrial processes, solar, exhaust gas, etc.), environmentally friendly refrigerant/absorbent solutions that contain no CFC or HCFC compounds to contribute to global warming or ozone depletion, and quiet, low vibration operation due to the absence of a mechanical compressor. Calculations based on electric production in the United Kingdom indicate that absorption chillers have only 67 to 81 percent of the global warming potential (GWP) and 31 to 36 percent of the acid emissions of electric motor-driven centrifugal chillers (Tozer 1994). These values may not be directly applicable to other areas of the world since the UK uses coal to produce a large proportion of its electricity, but it can provide an idea of the environmental savings that can be obtained. The use of alternate fuels and heat sources has an additional advantage in areas in which electric utility rates are high and peak demand charges can be a large part of the power bill. Some areas of the country experience brown-outs and rolling black-outs due to excessive demand in electric power. Air conditioning systems have been identified as a major factor in the growth of electric demand (Wilkinson 1994). It is in areas such as these that the absorption systems can have an overall LCC advantage over the base electric motor-driven, vapor-compression systems. One example of this advantage is shown for a generic application in the UK by Tozer (1994).

Data of the system in Figure 3 based on different temperatures can be adjusted to a common basis for comparison. Figures 5 and 6 show the COP and cooling capacity for a single-effect LiBr-H_2O absorption unit as functions of the temperature difference between the condenser (T_c, 80-110 °F) and the evaporator (T_e, 35-55 °F), with 100 °F at the absorber and 210 °F at the generator. As the graphs show, both parameters vary linearly with the temperature difference. The data was obtained using the SAM-15A system configuration in the ABSIM simulation software program (Herold and Radermacher 1995).

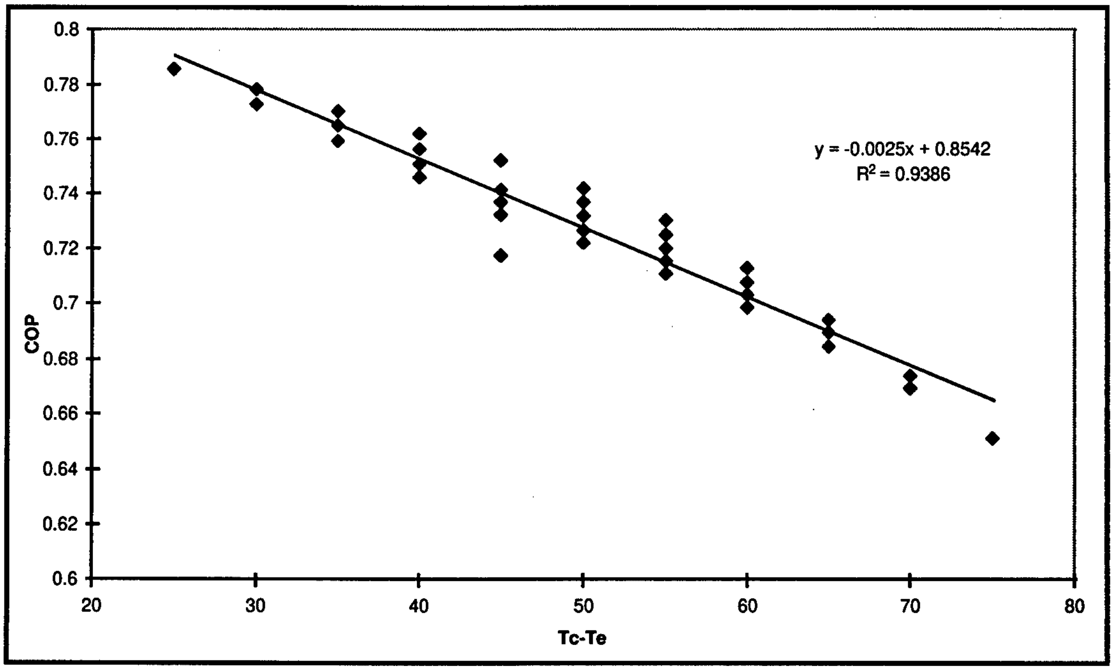

Figure 5. COP vs. condenser/evaporator temperature difference.

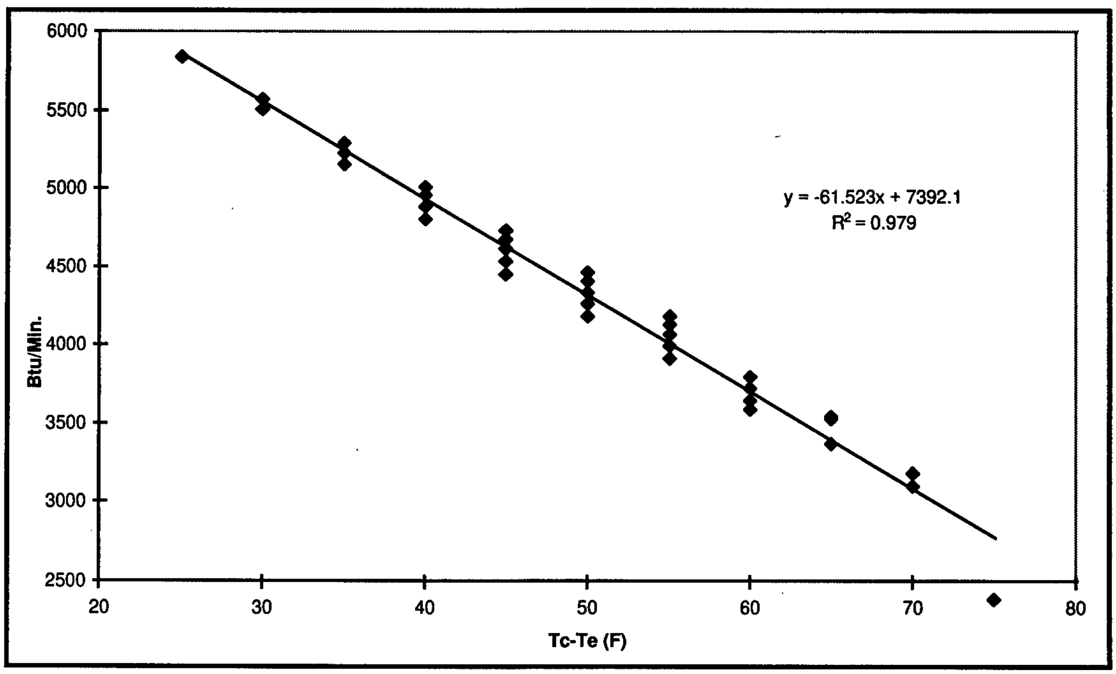

Figure 6. Cooling capacity vs. condenser/evaporator temperature difference.

Figure 7 shows the COP correction factor as a function of the evaporator/condenser temperature difference. The COP at a ΔT of 40 °F was given a value of unity. The COP values at other temperatures were then divided by the COP at a ΔT of 40 °F. The ΔT of 40 °F was chosen as the base since the ARI standards use T_c of 85 °F condenser water inlet temperature and T_e of 44 °F chilled water exit temperature. If the system under consideration is not rated at

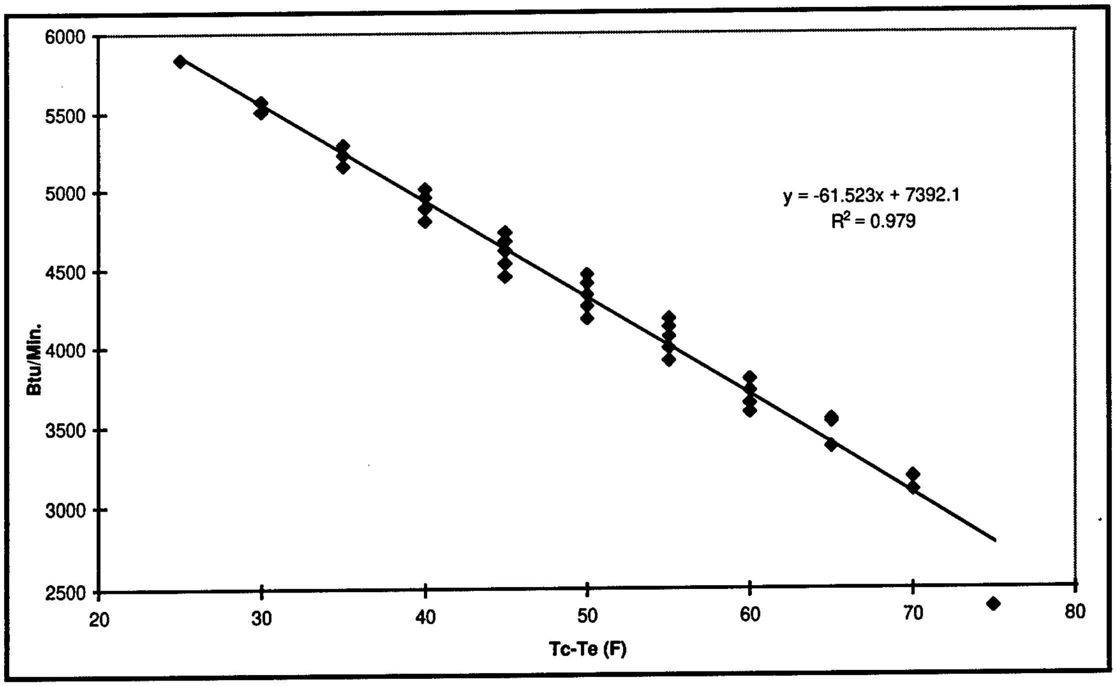

Figure 7. COP correction factor vs. condenser/evaporator temperature difference of an absorption chiller.

a ΔT of 40 °F, it can be corrected to the standard temperature difference by dividing by the proper correction factor for the given ΔT.

5.2 Operating Concerns

As one might expect, the differences of the absorption system bring with them some unique technical considerations for operation and maintenance. This section discusses some main aspects to be considered and examines the relation of COP to evaporator, condenser, and generator temperature (Ng et al. 1994).

One of the limitations of absorption systems stems from the use of a Li-Br salt solution as the absorbent. As the concentration of LiBr in the solution increases and the vapor pressure drops, there is an increased chance for the Li-Br salt to begin to crystallize. This limit imposes design restrictions on the evaporator, condenser, and especially the solution heat exchanger. The crystallization limit is dependent on the water vapor pressure.

As vapor pressure decreases, the LiBr concentration limit at a given solution temperature decreases. The fact that the evaporator temperature is dependent on the vapor pressure implies a lower limit on the chilled water temperature that can be obtained without crystallization. For typical system parameters, this limit is around 40 °F (Perez-Blanco 1993). Slightly lower temperatures are possible with the addition of crystallization inhibitors in the system.

The crystallization limits also have an impact on the liquid-to-liquid heat exchanger that is used to preheat the solution going to the generator. The solution coming from the generator has a relatively high mass concentration of LiBr (e.g., 65 percent). The pressure in the absorber is less than that in the generator, and the heat exchanger will lower the temperature of the returning strong solution. If the returning solution temperature is lowered too much in the heat exchanger, crystallization will begin to occur. This limits the amount of heat that can be recovered from the returning strong solution and thus also limits the performance capabilities of the system. The position in the system where crystallization is most likely to occur is the heat exchanger; low condensing pressures are conducive to crystallization. One method for controlling crystallization is to maintain artificially high condensing pressures even when low temperature cooling water is available (Stoecker and Jones 1980).

Double-effect chillers have additional parameters associated with the flow distribution between the two generator stages. Systems are called series flow if the LiBr solution flows from one generator to the next. They are called parallel flow if their is a split or distribution of the solution flow to the respective generators. The parallel flow systems typically offer better performance since the solution flow to each generator can be sized for the amount of heat input into the generator. The COP of a parallel flow type system varies with the distribution ratio as well as the concentration difference between the solution at the exit and entrance of the absorber. In general, an increase in concentration difference leads to an increase in COP. This can be seen intuitively by the fact that a high concentration difference means that an increased amount of refrigerant (water vapor) is recovered, which implies that an increased amount of the heat input from the generator is used for refrigeration. The optimum distribution ratio was determined to be about 0.28 for the concentration differences examined. When the solution distribution ratio was greater than 0.25, the possibility of crystallization decreased for the three concentration differences examined (Oh et al. 1994).

5.2.1 Problem Areas

Another consideration deals with corrosion of the components. Corrosion can occur inside the chiller due to the nature of the LiBr solution or on exterior components due to the heat source used to drive the system. As one might expect, the corrosive action of the LiBr solution increases with its temperature. In general, as the number of stages in an absorption system increases the temperature at the first generator also increases. This implies that special care must be used to combat corrosion in multiple-stage absorption systems. The corrosion rate for LiBr increases for temperatures at the first generator above 340 °F. The evaluation by Oh et al. (1994) showed that the generator can remain under this temperature limit if the distribution ratio is greater than 0.3 and the concentration difference is 4 percent or less. This is close to the optimum level for system COP and is also valid from the standpoint of crystallization discussed earlier.

Operation of numerous single-effect absorption machines at Pennsylvania State University has revealed that water quality is an important factor in the corrosion of components (Rose et al. 1995). The water in the region contains a great deal of dissolved carbonates due to limestone beds. Steam generated from this water contains carbon dioxide, which forms carbonic acid in the condensate. Acid corrosion of valves and fittings in the condensate return lines has been a major maintenance problem. A water treatment approach using an electrodialysis reversal process was installed in 1987 and has substantially reduced corrosion problems. Ineffective condensate removal from the generators can also compound the problem of corrosion as well as reduce system performance. A thorough evaluation of the water, steam, or heat source is necessary to determine if additional measures need to be taken to control corrosion and reduce maintenance costs.

5.2.2 Maintenance Needs

As noted earlier, absorption systems operate at pressures well below one standard atmosphere. The performance of the system is highly dependent on maintaining these low pressures. Therefore, any system pressure increase due to leakage of air into the system or the collection of non-condensable gases (NCG) must be controlled. These NCG typically collect in the low pressure section of the system (i.e., the absorber). The accumulation of these gases causes a partial pressure that is additive to the vapor pressure of the LiBr-H_2O solution. The evaporator temperature is dictated by the sum of these pressures (Murray 1993). As the pressure increases, so does the evaporator temperature. Although absorption systems are typically leak-checked by the manufacturer, it is virtually impossible to achieve a perfectly sealed system. Therefore, the greater the number of stages in a double-effect system, the more complex are the piping arrangements, leading to more valves and fittings that may act as leak paths for air to enter the system.

Another source for NCG is due to the reaction of the LiBr solution with the steel surfaces in the system. This reaction produces small amounts of hydrogen gas.

The double-effect systems will experience higher hydrogen generation rates due to the additional steel surface area in the system and the higher solution temperatures. After several months of operation, the chemical reaction of inhibitors nearly eliminates the generation of hydrogen in the absence of air in leakage (Murray 1993). Due to the potential performance problems associated with the build up of NCG in the system, all absorption machines have some sort of purge system to vent these NCG back to the atmosphere. Murray (1993) described various purge system arrangements. The most simple purge system consists of a heated palladium cell that can pass hydrogen gas to the atmosphere even though the hydrogen gas is at below 6mm Hg absolute. The disadvantage of this system is that it cannot pass nitrogen. As such periodic maintenance is needed to vent any accumulated nitrogen. Another system that was used on small residential units employed an auxiliary absorber, a fall tube, and a separator. Large commercial systems have more complex purge systems that typically use a vacuum pump to expel any NCG to the atmosphere. Over time the oil in these pumps can become diluted with condensed water, requiring maintenance of the pump to ensure consistent system performance (Murray 1993).

5.2.3 Rating

The chiller ratings shown in the collected manufacturer data are typically given at 100 percent design capacity. The units are rated according to ARI standards. The ARI standard specifies the condenser water inlet temperature (85 °F) and the chilled water exit temperature (44 °F). The actual unit capacity will vary as the above conditions change. The capacity and performance will decrease with higher condenser water or lower chilled water temperatures and vice-versa.

5.3 Energy Sources

5.3.1 Waste Heat or Cogeneration

There are various sources of heat that can be used to drive an absorption system. In general, the more stages in the system, the higher the required temperature at the first generator. In a single-effect LiBr-H_2O system, the optimum generator temperature is around 200 °F. As such, low-pressure steam (15 to 20 psig) is the most common heat source. Hot water (180 °F or higher) can also be used. The ability to use a low grade heat source makes the single-effect systems attractive when waste heat from processes or steam generation is available. Temperatures as low as 162 °F can be used, but capacity and COP drop with decreasing heat source temperature (American Gas Cooling Center 1994).

Double-effect systems typically require generator temperatures around 300 °F. These systems can be driven by high-pressure steam (nominally 120 psig or higher), be direct-fired with oil or natural gas burners, or use high temperature exhaust gas from another combustion process such as that from cogeneration. The two-stage units can operate at steam pressures as low as 45 psig, but at reduced capacity and performance (American Gas Cooling Center 1994).

5.3.2 Natural Gas

When examining gas-fired absorption systems, one has to be sure which heating value was used to determine the published fuel consumption. For example, most chillers that use natural gas are rated using the lower heating value (LHV) of natural gas (typically around 900 Btu/cu ft). However, natural gas is normally purchased and the charge by utilities is based on the higher heating value (HHV) (typically around 1,000 Btu/cu ft). To determine the fuel rate for estimating fuel costs, the published fuel consumption based on the LHV should be divided by the 0.9 LHV to HHV heating value ratio (EPRI 1992).

5.3.3 Solar Pond

The ability to use a variety of heat sources can lead to a wider method of application for the absorption system technology. A reverse application of absorption technology is to use it as a heat transformer for producing process heat from salt water solar ponds (Grossman 1991; Nielsen 1979). Figures 8 and 9 show a schematic of this type of arrangement. The typical temperature difference for the pond of 3.5 m deep is 86 °F (T_t) at the surface and 175 °F (T_b) at the bottom of the pond of 20 percent salinity. The temperature at the pond bottom is relatively low, but close to that for hot water-driven, single-stage, absorption systems, and the temperature at the top of the pond is at the typical condenser water inlet temperature. However, a temperature compatible to an absorption system can be produced in a heat transformer.

In the heat transformer operation, the bottom pond water is also circulated through the evaporator. Water from the absorption cycle is evaporated at this bottom temperature with the vapor sent to the absorber. The absorption of the vapor releases additional heat (the heat of absorption). This additional heat release provides a temperature lift to the solution. Water is circulated through an absorber heat exchanger and is heated to around 250 °F steam as it cools the absorber. This steam is available for use in an industrial process, or could be used as the driving heat source for an absorption chiller. The weak solution is sent to a recuperative heat exchanger and then to the desorber where water at T_b is used to drive off the water from the solution. This water vapor is sent to a

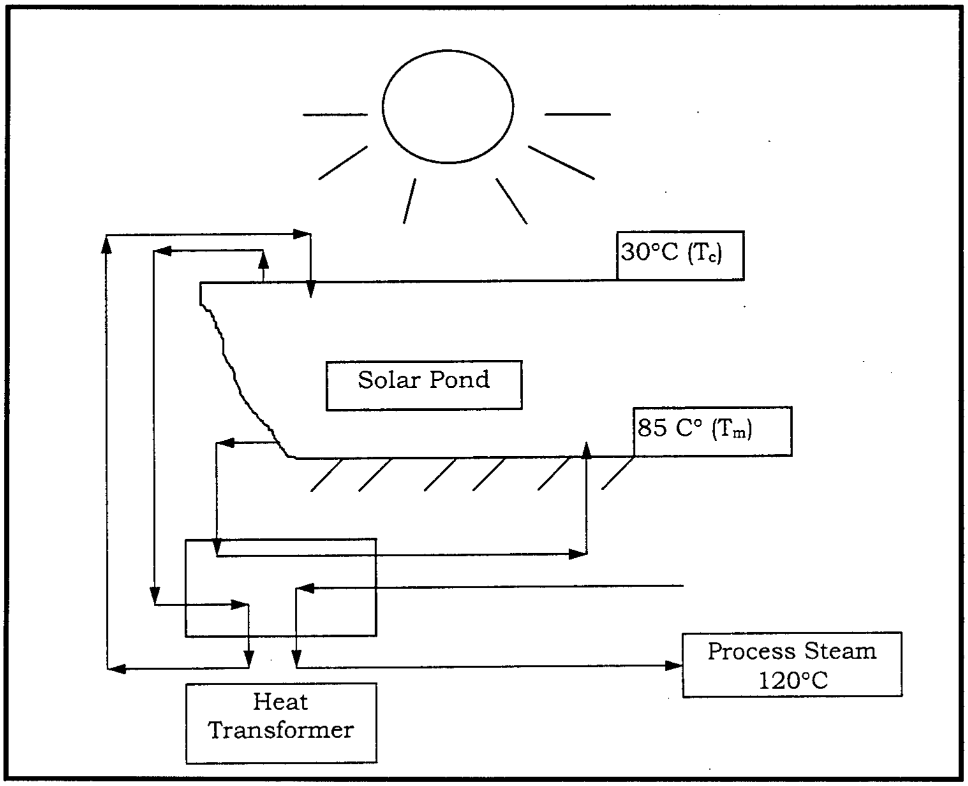

Figure 8. Solar pond heat transformer arrangement.

condenser where it is cooled and condensed by pond water at T_t. The calculated COP is about 0.5. The efficiency of conversion of solar radiation to process steam is about 10 times greater than that for electric power, assuming a Rankine heat engine operating at 5 percent efficiency (Grossman 1991). This could make the production of process steam more economically attractive than electricity depending on the relative costs of the two.

For example, in the case of a pond in the Chicago area (latitude 42 degrees), for which the yearly average of clear-day solar insolation is 0.2055 kW/m^2. Assuming cloudy days prevail for 25 percent of a year, the annual total available solar energy per acre is:

$$0.75\left(0.2055\, kW / m^2\right)\left(24\, hr / day\right)\left(365\, days / yr\right)\left(4{,}000\, m^2 / acre\right) =$$
$$5.4 \times 10^6\, kWh / yr / acre = 18.4 \times 10^9\, Btu / yr\, acre$$

At a pond efficiency of 55 percent, the total available heat production could be 10^{10} Btu/yr acre. The cost components per acre include:

- Land $4,000
- Construction $90,000
- Salt $100,000
- Total $194,000.

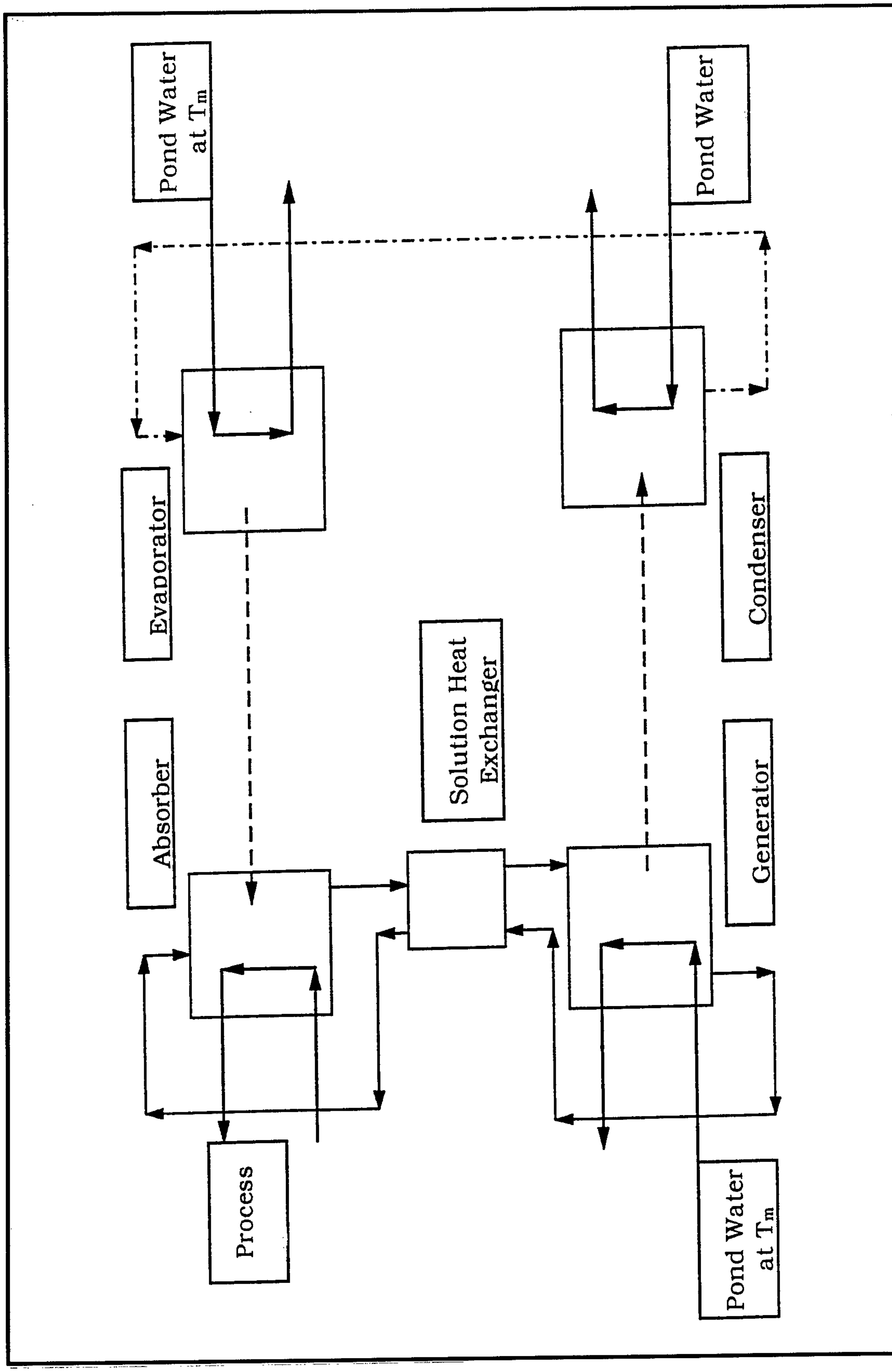

Figure 9. Heat transformer schematic.

The annual cost of owning and operating is nearly $30,000. This is because the temperature and concentration gradients in the pond have to be skillfully maintained. The value of the heat produced breaks even at $3/million Btu (Cha et al. 1980). This, when compared to that of the natural gas, cannot be considered encouraging. Hence the solar pond is not an economical method for absorption refrigeration.

5.3.4 Cogeneration

Another area in which absorption systems can provide potential economic benefit is in conjunction with cogeneration facilities. The high temperature exhaust or extracted high-pressure steam can be used to drive a double-effect absorption system. The formula for calculating the recoverable chilled water (tons) from the hot gas stream is

$$tons = (M)*(0.268)*(T_1 - T_2)*(0.97)*(COP)/(12{,}000) \qquad \textbf{Eq 19}$$

In the above relation, M is the pounds per hour mass flow rate of the exhaust gas; 0.268 is the average specific heat; T_1 is the inlet gas temperature; T_2 is the exhaust gas temperature leaving the absorption system heat exchanger; 0.97 is the estimated average losses from the ductwork; COP is the coefficient of performance of the absorption chiller; and 12,000 is a conversion from Btu/hr to tons (Hufford 1991). In many cogeneration systems, steam is typically extracted at around 15 psig to drive a single-stage absorption chiller. However, the analysis provided by Hufford (1991) indicates that, from a standpoint of overall equivalent energy recovery, it is more advantageous to extract steam at 115 psig to drive a parallel flow two-stage absorption system. This approach requires that only half the mass flow of steam is needed as compared to the low-pressure extraction to produce the same amount of cooling. The remaining steam goes all the way to a condensing turbine and produces additional electricity. The results of the comparison are shown in Table 10. It is assumed that electrical costs are $0.045/kWh; three alternatives are compared. The first is to extract steam at

Table 10. Energy equivalent of recoverable heat.

Energy Type	Alternative 1	Alternative 2	Alternative 3
Electricity Produced (kW)	192	608	1671
Chilled Water Equivalent Produced (kW)	2320	1122	0
Total kW Produced	2512	1730	1671

115 psig for a 2-stage absorption system. The second is to extract steam at 15 psig for a 1-stage absorption system. The third alternative is to use all the steam for the generation of electricity and to use an electric motor-driven chiller. This comparison assumes 20 lb/hr-ton of steam flow for the 1-stage, 9.7 lb/hr-ton of steam flow for the 2-stage, and 0.75 kW/ton for the electric motor-driven chiller.

5.4 Advanced Absorption Systems

There are a number of activities being pursued in the area of improved absorption chiller performance. These include optimizing the component design for the current 1- and 2-stage chillers, as well as more advanced 3- and 4-stage chillers offering even higher COP than currently available, but with increased complexity and cost. This section examines some of the potential future advanced absorption systems.

The first cost of an absorption chiller is generally higher than a comparable electric motor-driven chiller. Depending on the application and the relative energy costs, the operating costs of the absorption system can be noticeably lower than those of the electric system. For the current generation systems to become competitive, their first costs need to be reduced without sacrificing their performance. The main differences in first costs between absorption and centrifugal chillers are due to the additional heat exchangers required for the absorption system such as the absorber, generator, and solution heat exchanger (Herold 1995). Thermodynamic analysis of the absorption cycle provides insight into what areas performance enhancement efforts should be directed to obtain the highest possible gain. Aphornratana and Eames (1995) analyzed a single-effect absorption cycle using a second-law approach. The analysis takes into account the heat and work interactions in terms of the exergy, which increases during the irreversible processes. The magnitude of the exergy change provides an indication of the level of irreversibility in the process. This approach allowed them to quantify the irreversible processes in the various components of the system and thus provide an indication of where design optimization may give the greatest possibilities of improvements in performance.

By using this approach, they determined that a major source of losses are due to the irreversibilities in the heat transfer process. The components that account for the highest portion of irreversibilities are the absorber (20 to 40 percent of the total input) and the generator (28 to 35 percent of the total input). The exergy loss of the evaporator is much lower than the previous two components, but the evaporator has a higher impact on system performance. The irreversibilities of both of the first two components can be reduced by increasing the effectiveness of the respective heat exchangers and by reducing the circulation rate of the solution. However, higher heat exchanger effectiveness may lead to an increased possibility of crystallization given the same initial solution temperatures. The evaporator is identified as the component with the first priority for design change with the absorber being second. The second law analysis also supports the earlier statement that the single-effect absorption systems are more thermodynamically efficient in using low grade waste heat rather than high temperature heat sources (Aphornratana and Eames 1995).

The double-effect systems offer a higher COP as compared to the single-effect systems and are better suited for use of high grade heat sources. To further improve the use of high grade heat sources, a number of triple-effect systems are being evaluated. These systems include the three condenser-three desorber (3C3D) cycle, the double condenser coupled (DCC) cycle and the dual loop cycle. The 3C3D cycle is similar to the standard double-effect cycle, but with one additional stage. The configuration would resemble a double-effect system with the addition of a third condenser and a third desorber (generator). The DCC system uses heat from the high temperature condensers as an additional input to the low temperature generators. The dual loop cycle uses two single-effect loops in which heat from the condenser and absorber of one loop is used in the desorber of the second loop. Both loops provide a cooling effect.

Grossman et al. (1994) provided a description and comparative computer simulation analysis of these systems. Both series and parallel flow arrangements were considered for the three different cycles. The simulation showed COPs ranging from 1.272 for the series flow 3C3D cycle to 1.729 for the parallel flow DCC cycle. It further showed that the parallel flow arrangement for each cycle provided a higher COP with a lower risk of crystallization and slightly lower cooling capacity than the series flow arrangements (Grossman et al. 1994). As described earlier, the parallel flow arrangement provides this benefit since the amount of solution that passes through it can be matched to the capabilities of the components and their operating temperature characteristics. The triple-effect systems typically require a higher temperature at the externally heated generator.

In Grossman's simulation, the solution temperature leaving the first generator was set at 425 °F (the midpoint of what was determined to be the optimum range for a triple-effect system) design point. The Gas Research Institute has been working with an absorption chiller manufacturer to bring a triple-effect DCC cycle to market. They project COP in the range of 1.4 to 1.5 for their second prototype. Currently they are searching for interested parties for establishing a demonstration site in 1997. Their initial unit will be a 380-ton chiller. Their analysis has indicated that this system will be most economically effective in high load applications and can provide a substantial improvement with combined heating and cooling over conventional heater/chillers. The cost is expected to be about $200 per ton more than current double-effect systems and require about 25 percent more floor space (Ryan 1995).

Based on this information, it appears that triple-effect absorption systems will be viable options to meet the cooling needs of buildings. As these systems near market application, there is some effort in the area of four-effect systems. A computer simulation analysis has been carried out on a four-effect LiBr-H_2O absorption system (Grossman et al. 1995). The system design is an extension of the double-effect system using four condensers and four desorbers coupled together. Based on the earlier results of the triple-effect simulation, a parallel flow arrangement was chosen to provide the highest COP for the cycle. As with the triple-effect simulation, the component sizing (in terms of the overall heat transfer coefficient of the heat exchanger UA) for the computer simulation was based on SAM-15, a single-effect, solar-powered LiBr-H_2O system and the ABSIM computer code was used for the system modeling. The design point used a 600 °F temperature for the solution leaving the first externally heated generator. As noted in the triple-effect analysis, and shown in Table 11, the flow distribution between the four desorbers can have a noticeable impact on COP.

The design point COP was 2.013 (versus 1.729 for the triple-effect system). Noted that the simulation used properties from the 1985 ASHRAE Handbook for LiBr-H_2O solutions that were extrapolated beyond their stated range of validity. This was done since there was no data at the higher solution temperatures needed for a four-effect cycle. A complete detailed analysis and development of high temperature LiBr-H_2O solution properties needs to be conducted to further evaluate the performance as well as economic characteristics of this type of absorption cycle. Grossman et al. (1995) also pointed out some other challenges related to this type of system. Among these are the high flue losses associated with the lowered combustion efficiency to raise the firing temperature; this high temperature will accelerate and compound corrosion problems. Heat and mass transfer enhancement additives have a limited ability to survive at the elevated temperatures. (This is also a problem for the triple-effect systems.)

Table 11. Effect of desorber solution distribution on COP.

Unit 1 (lbs./min.)	Unit 2 (lbs./min.)	Unit 3 (lbs./min.)	Unit 4 (lbs./min.)	COP
5	5	15	35	1.556
10	15	15	20	1.925
15	15	15	15	2.013
20	15	15	10	2.075
30	10	10	10	2.137
35	10	7.5	7.5	2.167
40	10	5	5	2.177
45	5	5	5	2.153
35	15	5	5	2.165

6 Life Cycle Cost Analysis

An approximate LCC analysis can easily be done with many of the readily available computer spreadsheet programs. This section will establish the methodology used and cost factors to be considered. Later sections will provide sample LCC calculations comparing the base case and gas engine-driven vapor compression and absorption systems.

6.1 Net Present Value (NPV) Method

There are various methods for analyzing capital investment projects. This study will use the net present value method for calculating LCC. This method is preferred since it takes into account all relevant cash flows as well as the timing of those cash flows. The analysis begins by establishing the relevant cash flows associated with the project. Relevant cash flows include those costs that would be incurred from today if the specific project were chosen. Previously incurred (sunk) costs should not be included in the present analysis as the current specific decision cannot affect past expenditures. These relevant costs can be broken down into the initial costs (e.g., purchase, shipping, installation, etc.), operational costs (maintenance, fuel costs, etc.), and termination costs (salvage value, site cleanup, etc.). For the purposes of this study, the termination costs are assumed to be negligible. This study also assumes that, for a given application, the choice of systems will be mutually exclusive (i.e., only one alternative can be chosen). To simplify the calculations, all cash flows are assumed to occur at the end of the operational year.

Once a system has been sized, the initial costs can be determined from the system matrix information or from conversations with representatives from equipment manufacturers. The latter of the three may help to provide a more accurate assessment of installation costs for the specific application than the others. As can be seen from the manufacturers' data, electric motor-driven chillers typically have a lower purchase cost than both the gas engine-driven and absorption systems. Another factor to be considered in the total first cost is the additional cost for installation of the system on site. This can include site preparation, piping, labor costs, and ancillary equipment such as pumps and cooling towers if needed. This LCC analysis considers a 15-year investment horizon. There are no provisions for major equipment replacement or overhaul in that time frame.

6.2 Purchase Cost

Figures 10 and 11 show the purchase cost information for the base case systems. These curves show the relation of purchase cost as a function of ton rating. The systems were divided into air-cooled and water-cooled condenser units since this was the major factor differentiating the costs between systems of similar ton ratings. The curves are provided here for comparison to the cost curves for the engine-driven vapor compression and absorption systems that are discussed and compared in the following sections.

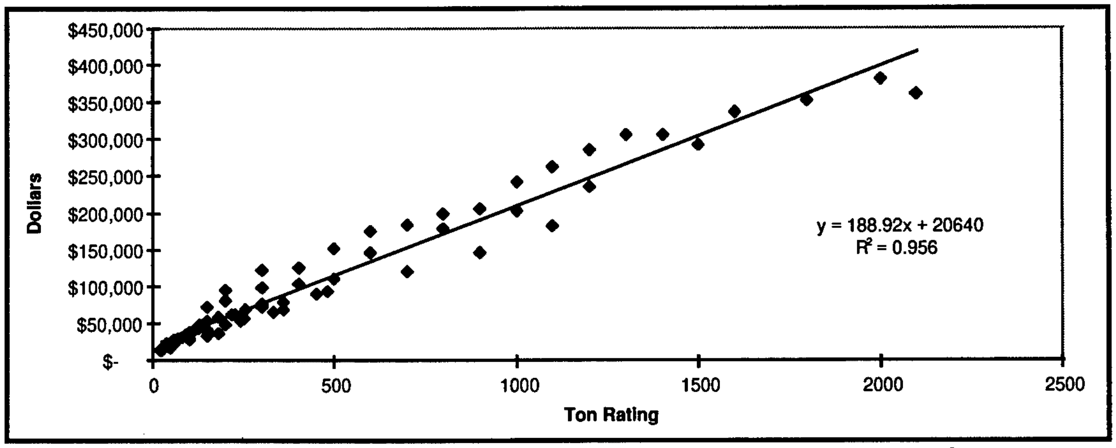

Figure 10. Purchase cost vs. ton rating for motor driven vapor compression systems with water cooled condensers.

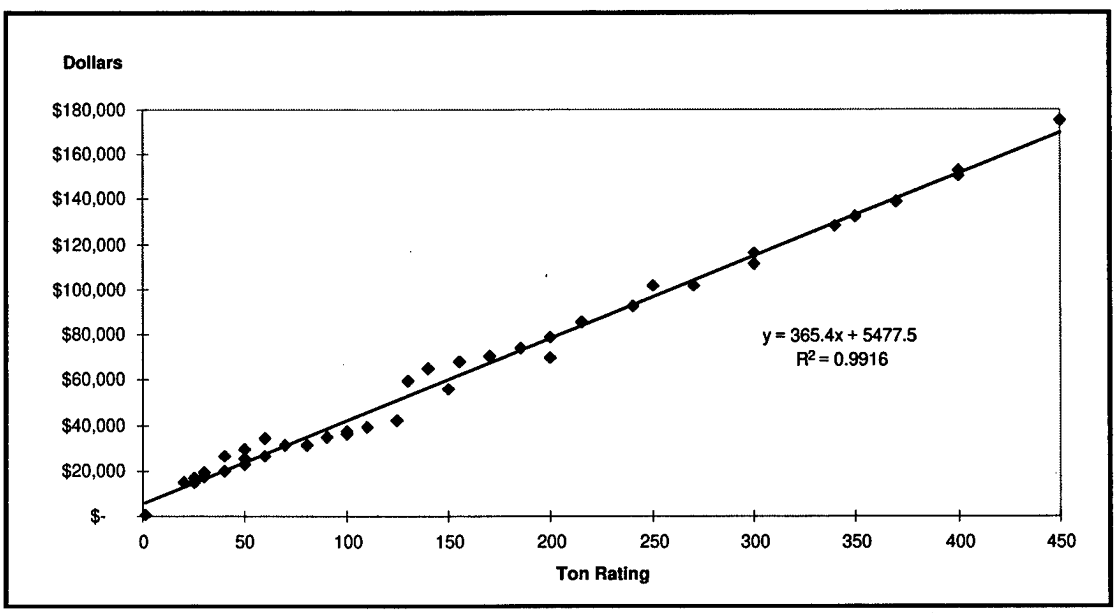

Figure 11. Purchase cost vs. ton rating for motor driven vapor compression systems with air cooled condensers.

6.3 NPV Calculation

A good way to evaluate a project from a financial standpoint is to use the Net Present Value (NPV = LCC + first cost). This takes into account all the cash flows as well as the timing of those cash flows. Since the purchase and operation of either the vapor compression or absorption system results in only cash outflows, the NPV for these two options will both be negative. The least negative value represents the best option. This is the system that will have the least impact on the total assets of the owner.

6.4 Energy Costs

This study focuses on estimation of annual energy costs as the input for the operational cash flows. Annual energy costs depend on the type of energy used (gas or electric), the specific utility rate structure, the chiller efficiency, and the annual cooling demand. Operational costs will typically comprise the majority of cash flows for a given project. For a cooling system, these operating cash flows are primarily due to energy costs. Maintenance costs may also be a major factor depending on the type of system, age of equipment, and specific application requirements. This section describes a method for the estimation of annual energy demand of air-conditioning and heat pump systems for housing facilities or supplementary or direct heating at a given geographical location (as distinct from heat generating facilities such as manufacturing or central computing facilities). This methodology, when developed, will allow for the comparison of various options in the design of an installation by estimating the annual energy cost of an installation, determining the balance between cooling and heating requirements, choosing the equipment and heat source or sink, and choosing the capacity and number of units. The primary data is the demand due to cooling degree days (CDD) or heating degree days (HDD) (Huang et al. 1987) and their modification due to internal gain, ventilation, humidity, and insolation (Claridge et al. 1987).

These HDD and CDD have been defined in a variety of ways according to specific concerns other than temperatures. The degree days are defined primarily for family housing, but can be extended to office buildings and barracks.

1. Daily HDD at base temperature from:
   46 to 65 °F – 30-year average published by NOAA.
   Daily CDD at base temperature from:
   46 to 65 °F – NOAA.

2. Hourly heating degree days (HDHR) at base temperature:
   46 to 65 °F
   heating degree hours divided by 24.

   Hourly cooling degree days (CDHR) at base temperature:
   60 to 79 °F
   cooling degree hours divided by 24.

   (Weather tapes are available for the above data.)

3. Vented hourly cooling degree days (CDHRV) at base from:
   60 to 78 °F
   and humidity ratio greater than 0.0116 lb
   water vapor per lb of dry air.

4. Heating insolation days (HID) at:
   57, 60, and 65 °F base temperatures to correct for solar input.

   Cooling insolation days (CID) at:
   60, 65, and 70 °F.

   Also vented cooling insolation days (CIDV).

For instance, at a base temperature of 65 °F, the meaning of the above database can be seen for a few typical geographical locations in Table 12. Table 13 shows a simplified comparison of different cooling degree values based on different calculation methods for several different locations. The analysis will use the base NOAA CDD as well as using the cooling degree day with insolation and ventilation (CIDV) as this should provide a good range for comparison.

Table 12. Degree days designations at different geographical locations (65 °F base temperature).

Parameter	Portland, ME	Chicago, IL	Fort Worth, TX	Miami, FL
HDD	7537	6065	2229	142
HDHR	7714	6238	2555	185
NOAAHDD	7501	6177	2407	119
HID	7441	6283	4116	499
Winter ID(mon.)	5258(7)	4866(7)	3433(4)	695(1)
CDD	292	713	2500	4180
CDHR	504	865	2689	4227
NOAACDD	254	955	2809	4095
CDHRV	303	634	2436	4019
CID	4183	4845	8762	12229
CIDV	1820	2741	6958	11191

Table 13. Comparison of cooling degree day values (65°F base temp.).

Parameter	Portland, ME	Chicago, IL	Ft. Worth, TX	Miami, FL
CDD	292	713	2500	4180
CDHR	504	865	2689	4227
CDHRV	303	634	2436	4019
CID	4183	4845	8762	12229
CIDV	1820	2741	6958	11191

Insolation is seen as a significant factor in small family housing. The CDD and HDD applicable at a given location and type of housing remain to be validated for specific applications. One notes that, for a moderately sized office building, the valid reference might be HDD and CIDV (to account for air change) and the choice of equipment might be:

- Portland, ME — Furnace heating (large) + AC
- Chicago, IL — Furnace heating + HP/AC
- Fort Worth, TX — HP/AC
- Miami, FL — Furnace or electric heating (small) + AC (large).

The annual electric demand for cooling by an electric motor-driven system is given by:

$$kWh_c / yr = 84.38 \int_{yr} \left[N_c / \left(COP_c \right)_e \right] dt = 84.38 \sum_{yr} \left[N_c / \left(COP_c \right)_e \right] \left(operating\,hrs / day \right) / 24 \qquad \text{Eq 20}$$

in which N_c is the cooling capacity in tons of ice/day, and $(COP_c)_e$ is the coefficient of performance for cooling based on electric work. Instantaneous values of N_c and $(COP_c)_e$ are functions of T_{amb} and $T_{chiller}$, the ambient and the chiller temperatures. For a given design of chosen equipment with maximum capacity N_{co} and corresponding $(COP_c)_{eo}$, for $\Delta T_{co} = T_{amb} - T_{chiller}$; in which subscript "o" denotes design values corresponding to reference ambient and chiller temperatures, the above relation can be approximated by:

$$kWh_c / yr \cong 84.38 \left[N_{co} / (LF)(COP_c)_{eo} \right] \left[(CDD / \Delta T_{co}) \right] C_1 \qquad \text{Eq 21}$$

in which CDD is the cooling degree days per year at the given location. The accuracy of this approximation remains to be validated by comparing results of Equations 20 and 21 to that of a few actual cases. C_1 is a correction factor to be determined for a specific system to account for operating conditions (WGNAS [1993] introduced this as a CDD factor of 1.96); and the load factor (LF) corrects for reduced motor efficiency at part load (or loss in on-off operation) or variable speed drive, idling with intake-valve-unloader or suction unloader on compressor, and other control modes. EPRI [1992] gave the range of 0.80 to 0.91 and for an actual case, it was determined to be 0.784.) When EER is used in place of COP, the substitution is EER/3.413 for COP.

The same system can be operated as a heat pump at a location where frost does not occur, at $\Delta T_h = T_h - T_{amb}$, in which T_h is the temperature at which heating is carried out. If these temperatures correspond to the upper and lower values of ΔT_{co}, or $\Delta T_{ho} = \Delta T_{co}$, a reference temperature difference, the maximum attainable heat pump output in tons N_{hpo} or 12,000 N_{hpo} Btu/hr is given by:

$$N_{hpo} = N_{co} + W = N_{co} + N_{co} / \left(COP_c\right)_{eo} \qquad \textbf{Eq 22}$$

with W being the corresponding electric work input, and:

$$\left(COP_h\right)_{eo} = \left(COP_c\right)_{eo} + 1 \qquad \textbf{Eq 23}$$

These relations give:

$$N_{hpo} / \left(COP_h\right)_{eo} = N_{co} / \left(COP_c\right)_{eo} \qquad \textbf{Eq 24}$$

A similar approximation as in Equation 21 gives, for heating degree days HDD:

$$kWh_h / yr \cong 84.38\left[N_{hpo} / (LF)\left(COP_h\right)_{eo}\right]\left[(HDD) / \Delta T_{ho}\right]C_2 \qquad \textbf{Eq 25}$$

The maximum heating capacity given by 12,000 N_{hpo} Btu/hr might be greater or smaller than that needed at the given location depending on its particular heat loss pattern and winter weather. In that case, actual needed $[N_{hpo}/(COP_h)_{eo}]$ should be used. C_2 is a correction factor including those of C_1, with reduced insolation, humidity correction, etc. (Claridge et al. 1987). In case of frosting over a duration to give a heating degree days with frost of HDD_f when actual $T_{amb.}$ is below frosting:

$$\begin{aligned} kWh_h / yr \cong 84.38\Big\{&\left[N_{hpo} / \left(COP_h\right)_{eo}\right]\left[\left(HDD - HDD_f\right) / \Delta T_{ho}\right] \\ &+ \left[N_{hpo}\right]\left[\left(HDD_f\right) / \Delta T_{ho}\right]\Big\}C_2 \end{aligned} \qquad \textbf{Eq 26}$$

when defrost resistance heating has to be used, either in the form of defrosting coils on the evaporator or simple resistance heating as supplement when actual maximum demand for heating is greater than 12,000 N_{hpo} Btu/hr. If this maximum rating is much higher than the actual demand, an option is to have only a reduced number of machines operated as heat pumps.

For absorption systems or engine-driven systems, the energy demand in million Btu/yr is given by replacing the quantity 84.38 $N/(COP)_e$ with $0.288N/(COP)_s$ for refrigeration or heat pump. Note that a big factor in the electrical cost and the resulting LCC is the demand charge on an electric-driven system. Where a chilled water storage tank can be accommodated (Sohn 1990), an electrical powered system has a definite economic advantage.

An exception to the above considerations is seen in heat generating manufacturing facilities and large computing centers where heat has to be removed all year long (Nault et al. 1995). The same reference demonstrated that for a manufacturing facility of the National Semiconductor Corporation in Portland, ME, free cooling can be applied. Also, in the course of a CFC replacement program, when the ambient wet bulb temperature is below 39 °F, the heat exchanger transfers heat from the returning chilled water to the water returning to the cooling tower. They managed to reduce the system capacity from 7,500 tons in 1990 to 6,180 tons in 1994 with HCFC-22 replacement, and also to reduce the total operating hours of refrigeration from 49,948 hours to 27,207 hours per year in 1994 with accrued saving of energy consumption of 9,792,200 kWh/yr and reduced energy cost of $695,000/yr.

6.5 LCC Comparison of Engine-Driven Vapor Compression to Base Case

Gas engine-driven cooling equipment has been suggested by many to offer large savings in operating costs over the life of the equipment as compared to equivalent vapor compression systems. Life-cycle cost comparisons can be made to gain an insight into the benefits of gas engine-driven equipment as compared to conventional vapor compression cycle equipment. The cost components include initial purchase price, installation, maintenance, and operation.

Pricing was collected from manufacturers of gas engine-driven chillers. These quotes provide an extensive database for estimating purchase costs of various systems and provides a starting point from which to do life cycle cost analyses. Actual quoted costs for systems from different manufacturers varied at a given tonnage. Figure 12 shows the relation of total cost to ton rating for water-cooled engine-driven chiller systems. Figure 13 illustrates the relation of total cost to ton rating for air-cooled, engine-driven chiller systems.

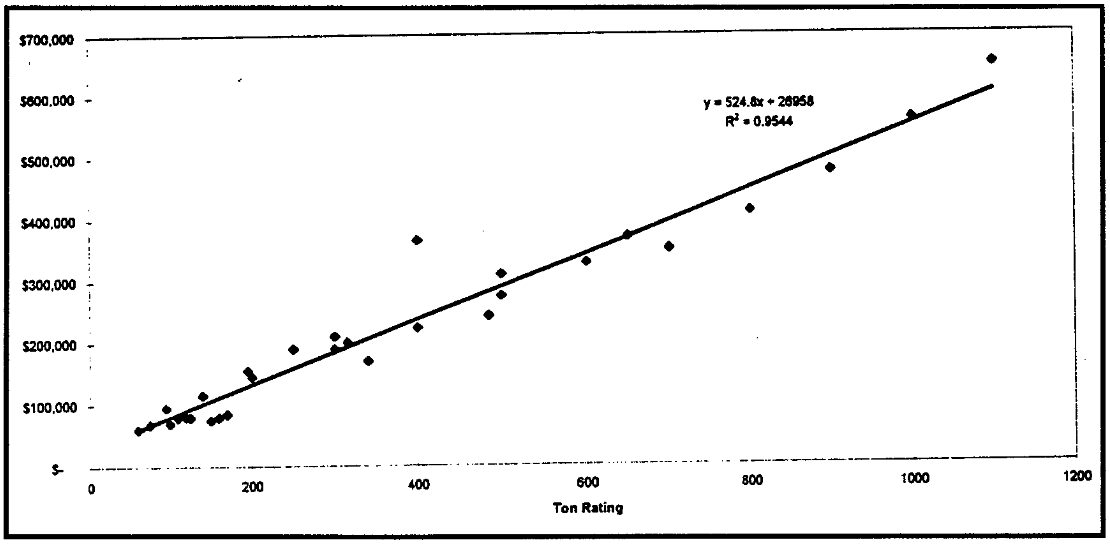

Figure 12. Relationship of initial cost to ton rating for water-cooled gas engine driven chillers.

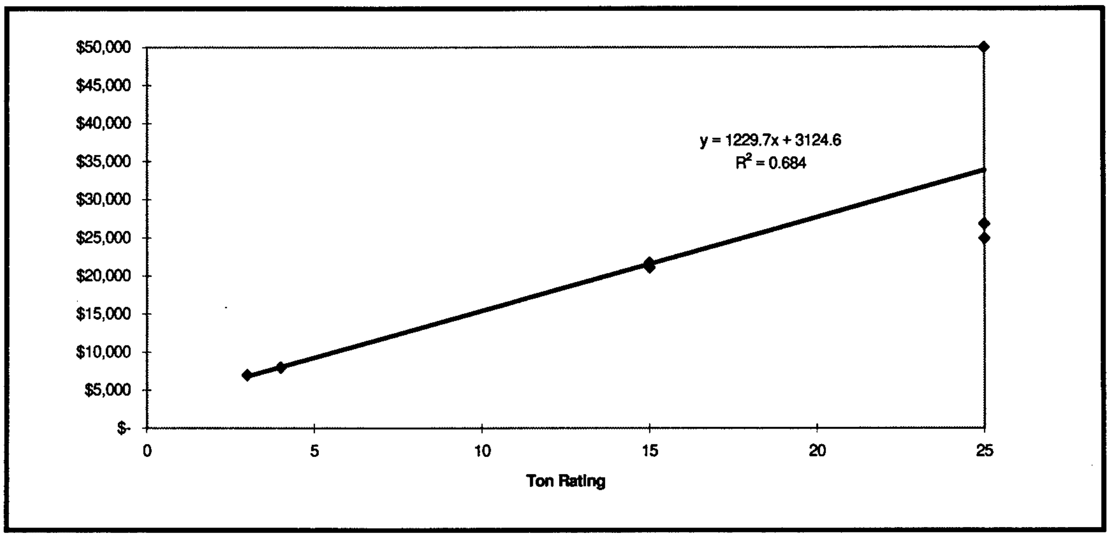

Figure 13. Relationship of initial cost to ton rating for air-cooled gas engine driven chillers.

Installation costs are very difficult to measure for new cooling systems. Costs are entirely dependent on the characteristics of the locations where the equipment is to be installed. Factors affecting installation cost include time and labor requirements, labor rates, piping, and control options, among others. Manufacturers are therefore very reluctant to include installation information when quoting budget prices. Cler (1995) reported some estimates of engine-driven installation costs. The report CFCs and Electric Chillers by the Electric Power Research Institute (EPRI 1992) includes approximations of installed costs of gas engine-driven chillers. Contractors and records of previous projects also represent good sources of information regarding both installation and maintenance costs of gas engine-driven equipment.

The main cash flows from the investments in cooling equipment are the operating costs. Operating costs include maintenance and energy costs. Energy costs for engine-driven systems consist primarily of the cost of natural gas. Natural gas costs can vary considerably between locations and can also vary seasonally. Gas engine-driven chillers are most competitive in areas in which electric rates are high and natural gas rates are comparatively low. Incentives may exist in some areas for the use of gas cooling equipment that could make the life cycle cost analysis of gas engine-driven systems more favorable.

A life cycle cost analysis comparing an electric motor-driven unit to a gas engine-driven unit was performed for the Willow Grove Naval Air Station (1993). The analysis was done for a 15-ton chiller. Characteristics of the air-cooled electric motor and gas engine-driven units are provided in Tables 14 and 15. Equivalently sized units were studied for comparison purposes. Cooling degree days for WGNAS are 946 (WGNAS 1993). A conversion factor of 1.96 was applied to this value to obtain equivalent cooling degree days (cooling degree days with insolation). The efficiency factor and annual maintenance approximation were obtained from EPRI (1992). Electrical and demand charges were obtained from the utility rate structure of WGNAS.

Two cases were considered in the life cycle cost analysis. One analysis was made in which data obtained in the course of this study were used to do a life-cycle cost analysis for an electric motor-driven chiller. A second analysis was made in which data from WGNAS was used to do a life cycle cost analysis. Table 15 summarizes the results. A similar procedure was conducted for the gas engine-driven chiller. Study results refer to approximate values defined in the course of this study. Willow Grove results refer to actual historical data obtained from Willow Grove documentation. The electric charge was calculated by multiplying the number of kWh/yr by the electrical cost of $0.03/kWh. The number of kWh/yr for the present study results was calculated by using the formula that kWh/yr is equal to:

$$(84.38)(tonnage)(equiv.CDD)/(eff.factor)(COP_0)(\Delta T_0) \qquad \textbf{Eq 27}$$

Table 14. Characteristics of the electric motor driven chiller utilized for comparison.

Chiller characteristics	Value
Ton rating	15
ΔT_0	50°
Equivalent CDD/yr	1,854
$(COP_c)_e$	3
Efficiency factor	0.8
Maintenance	$100/yr
Electrical cost	$0.03/kWh
Demand charge	$23.70/kW month

Table 15. Cost comparison for electric motor driven chillers.

Parameter	Study Results	WGNAS Report
Electric charge	$587	$1,038
Auxiliary (15% of compressor drive)	$88	(included)**
Demand charge	$2,560	$3,412*
Auxiliary demand charge	$212	(included)**
Maintenance	$100	(included)**
Total operating	$3,549	$4,450
Present value (15 yr., 4% discount rate)	$39,459	$49,477
Annual cost of owning and operating	$4,989	$7,872
Initial cost (installed)	$21,600	$51,330
Net present value	-$61,059	-$100,807

* Original WGNAS estimates gave $14,379, which appears to be due to a demand charge of nearly $100/kW mon., rather than $23.70/kW mon.
**These values are included in the electric charge of $1,038.

This formula approximates the annual demand for cooling by an electric motor-driven system based on equivalent cooling degree days. WGNAS was found to use approximately 19,557 kWh/yr. The report stated that 34,618 kWh were actually measured. Information from the report indicated that the electric chiller had a 22.5 kW motor including an auxiliary drive. However, calculations based on the kW/ton ratio indicate that an 18 kW motor would suffice. The oversized motor could have resulted in the excessive number of kWh measured. A demand charge of $23.70/kW-month was stated in the report and was used in the analysis for each case. The total operating charge includes both annual maintenance and energy cash flows. The present value, for 15-year life and 4 percent discount rate with annual charges beginning at the end of the first year of purchase, is given by multiplying the annual operating cost by an LCC multiplier (LCCM):

$$
\begin{aligned}
LCCM &= (1.04)^{-1} + (1.04)^{-2} + \ldots + (1.04)^{-15} + (1.04)^{-16} \\
&= \left(1 - 1.04^{-15}\right) / 1.04\left(1 - 10.4^{-1}\right) = 11.1184
\end{aligned}
$$

(Note that WGNAS (1993) used a multiplier of 11.85 for the life and discount rate; the reason for this is unclear.) Alternatively, this multiplier can be found by referring to annuity tables. The net present value includes the present value of the annual operating cost cash flows and the initial cost of the machine. Therefore, all relevant cash flows are discounted back to time zero (the purchase and beginning of operation of the machine). This allows the results of the two cases to be compared to one another as well as to the net present value results from the gas engine-driven chiller analysis. The annual cost of owning and operating the equipment was found by adding the total annual operating charge to the value of the annual depreciation of the equipment, assuming straight-line depreciation over a period of 15 years.

A similar analysis was also made for gas engine-driven chillers. Table 16 identifies characteristics of the gas engine-driven chiller. Table 17 summarizes the results of the analysis. The natural gas charge was calculated by multiplying the number of MMBtu/yr by the natural gas price as stated in the report of \$6.50/MMBtu. The number of MMBtu/yr for the study results was calculated by using the formula:

$$MMBtu\,/\,yr = (288{,}000)(tonnage)(equiv.CDD)/(10^{6})(COP)(\Delta T_{o}) \qquad \text{Eq 28}$$

This formula approximates the annual demand for cooling by an engine-driven system based on equivalent cooling degree days. It gave 200 MMBtu/yr as compared to 387.5 MMBtu/yr recorded at WGNAS, probably because of an idling condition of the engine or the choice of equivalent CDD. The saving LCC is the difference in present values between the electric motor and gas engine-driven systems for each of the cases. The payback period was calculated by dividing the first cost difference between the two systems by the difference in total annual operating charge. The comparison between the electric and gas engine chillers is less favorable to the gas engine-driven system in terms of net present value (NPV) in each of the two cases.

Table 16. Characteristics of the gas engine driven chiller utilized for comparison.

Chiller Characteristics	Measure
Ton rating	15
ΔT_o	50°
Equivalent CDD/yr	1,854
$(COP_c)_e$	0.8
Maintenance	\$500/yr
Electrical cost	\$0.03/kWh
Natural gas price	\$6.50/MMBtu

Table 17. Cost comparison for engine driven chillers.

Parameter	Study Results	WGNAS
Natural gas charge	\$1,300	\$2,519
Electric charge	\$100	(included)
Maintenance	\$500	(included)
Total operating	\$1,900	\$2,519
Present value (15 yr., 4% discount rate)	\$21,125	\$28,007
Saving LCC	\$18,334	\$21,470
First cost difference	\$23,400	\$26,642
Payback period	14 years	14 years
Annual cost of owning and operating	\$4,900	\$7,717
Net present value	-\$66,125	-\$105,979

Table 18. Required break even electric charges for a gas price of $6.50/MMBtu.

Break-Even Electrical Cost	
Study Results	**WGNAS**
$0.02996/kWh	$0.02548/kWh

The gas engine-driven system is not attractive under the conditions of this analysis unless an exorbitant demand charge is collected by the utility. The high first cost of the gas engine-driven system greatly contributes to the unfavorable net present value. Table 18 provides the electric rates for each case that would allow the electric motor-driven system to break even with gas engine-driven system based on the annual energy expense disregarding the demand charge.

Note that, based on the equivalent CDD of 946x1.96 = 1,854 given in WGNAS (1993), the above analysis favors the engine-driven system based on annual operating charge, but gives much lower energy needs (19,557 versus 34,618 kWh/yr of electric power; 200 versus 387.5 MMBtu/yr of gas) than those recorded for WGNAS. However, if the CIDV of the Philadelphia area (close to WGNAS) of 4,041 degree days per year (Huang et al. 1987) were used (CDD = 1,081, close to 946 of WGNAS), the energy needs would be 42,633 kWh and 387.9 MMBtu/yr respectively. The higher value of degree days will make the engine-driven unit less favorable than in the above estimates because the demand charge for the electric chiller option remains unchanged. Further confirmation of the valid procedure will be desirable.

The NPV represents the total payout over the lifetime of the systems.

6.6 LCC Comparison of Absorption System to Base Case

This examines the specific cost components of absorption systems and how they compare to those of electric motor-driven vapor compression systems. Previous LCC comparisons of these two systems will be examined and the EPRI results (1992) will be compared to results obtained using information developed in this study. Cost components include purchase, installation, maintenance, and operating costs.

6.6.1 Purchase Cost

The manufacturers' information that was collected provides an extensive database for estimating the purchase cost of various systems. As may be expected, the actual costs vary among manufacturers for systems of the same tonnage. Figure 14 shows the relation of purchase cost to ton rating for single-effect absorption systems. These are all indirect (steam) fired units. Figures 15 and 16 show the same relation for double-effect absorption units. The double-effect systems are divided between indirect (steam, Figure 15) and direct-fired (Figure 16) systems. Most manufacturers noted that, above approximately ratings of 1,000 tons, the cost per ton becomes relatively constant. Figure 17 shows an alternate view of the purchase costs for double-effect absorption systems. This chart shows the high, low, and average price per ton for four different ton rating levels. This chart gives an indication of the range of costs for different rated systems and also has the characteristic of relatively constant cost per ton at higher ton ratings. The ton ratings were chosen based on the availability of sufficient data points at the particular ton rating level. A similar chart for 1-effect systems could not be developed due to lack of sufficient data points at particular ton ratings.

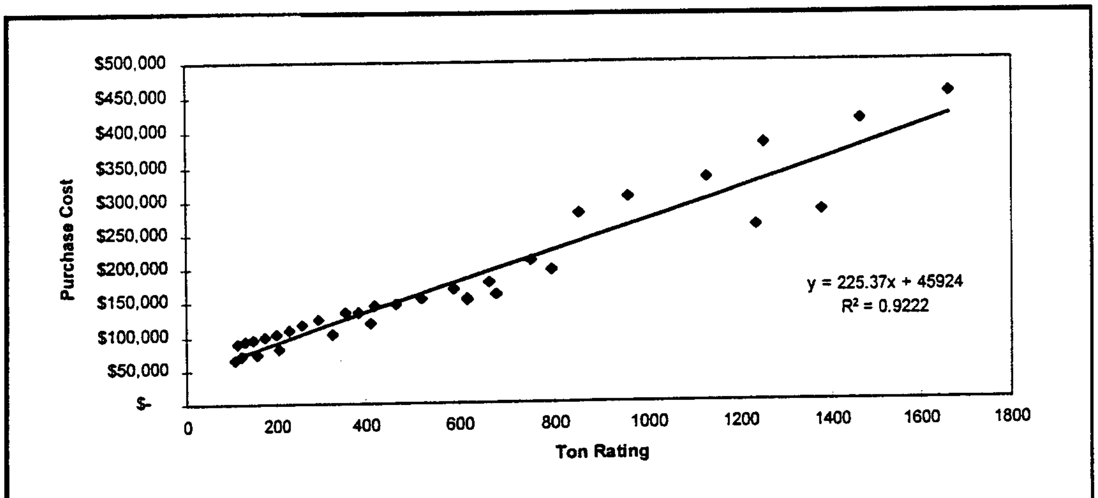

Figure 14. Purchase cost vs. ton rating for 1-effect absorption systems.

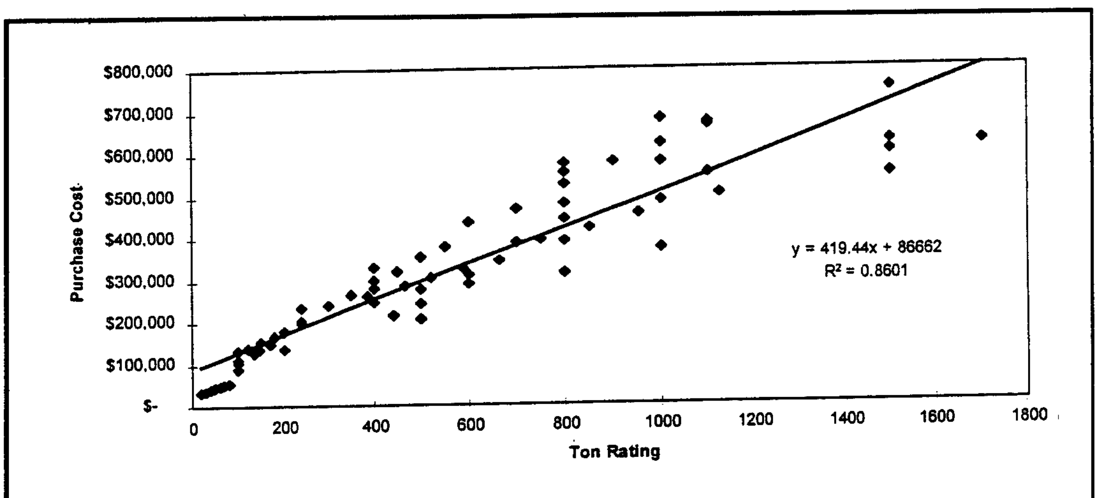

Figure 15. Purchase cost vs. ton rating for indirect fired 2-effect absorption systems.

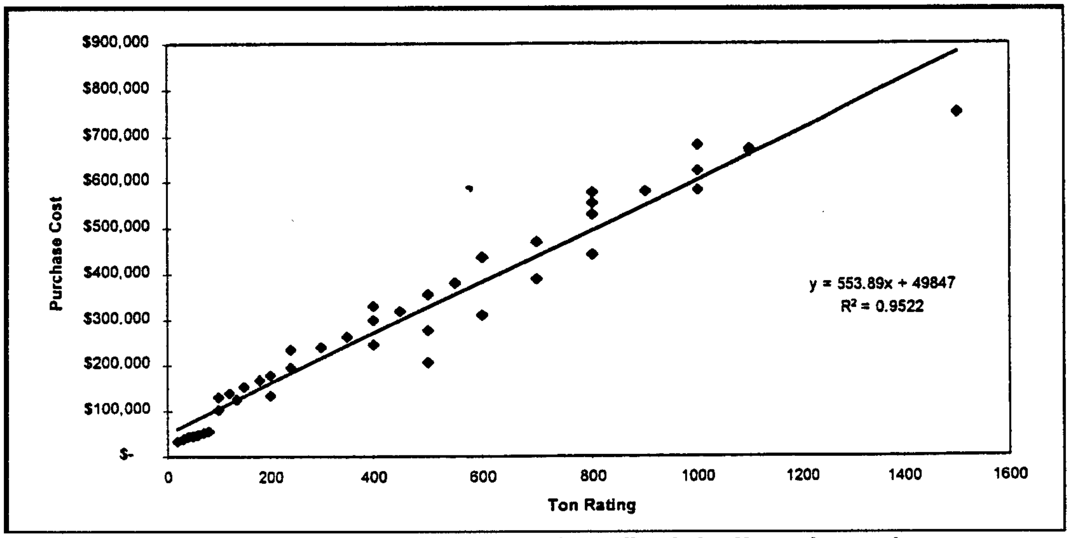

Figure 16. Purchase cost vs. ton rating for direct fired, 2-effect absorption systems.

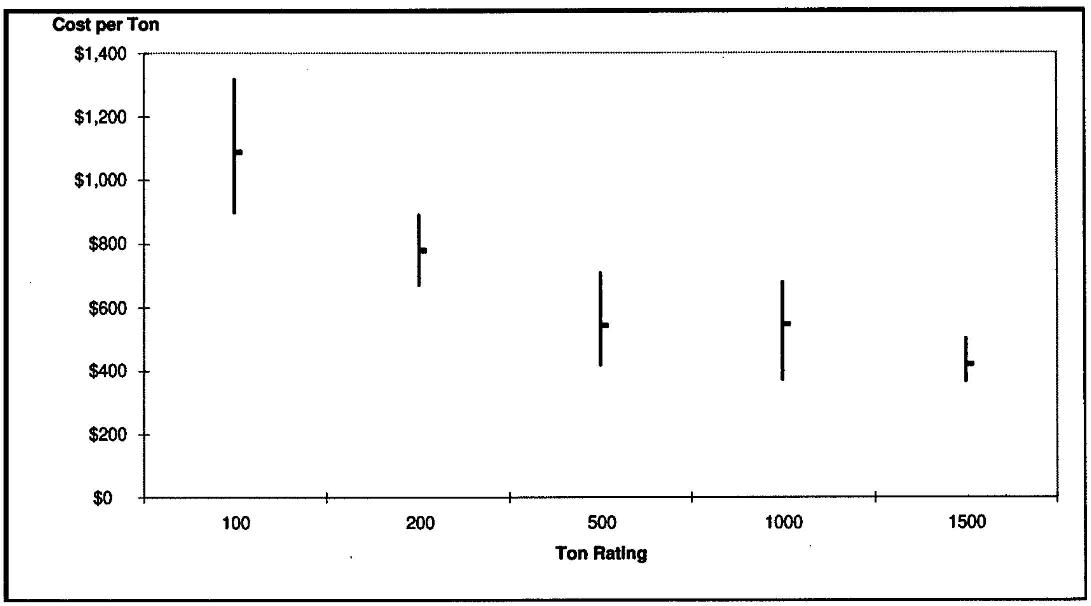

Figure 17. Range of cost per ton vs. ton rating for 2-effect absorption systems.

The other cost factor to be considered with the purchase price is the cost for delivery and installation. Manufacturers typically quote installation costs on a case-by-case basis since the particular system and building requirements can have a large impact on the actual cost. Location, labor rates, piping, and control options can all have an impact on installation costs. Review of various reports (EPRI 1992; Cler 1995) and conversations with manufacturers indicate that a typical rough estimate for installation costs is approximately 50 percent of the chiller purchase cost. This can be used for budgetary comparison of different systems. However, a final decision should be based on financial analysis using actual quotes from manufacturers.

As we mentioned earlier, the main cash flow requirements are due to operating costs (energy, maintenance, etc.). For absorption systems, the energy costs are typically the cost of natural gas for the direct-fired units or the marginal costs associated with added steam production (if any) for the indirect-fired units.

Direct energy costs vary considerably from region to region and should be determined on a case-by-case basis with the local utility supply company. In situations where waste heat is available either from industrial processes or cogeneration facilities, the direct energy costs can be considered negligible. It is in these situations where absorption systems have their greatest LCC advantage. Maintenance costs can also vary depending on operating cycle and water quality. Many suppliers will provide maintenance contracts based on the operating requirements. These can be priced on a dollar per ton-year or dollar per ton-hour basis. Cler (1995) reports typical maintenance costs for absorption systems in the range of $0.007 - $0.009 per ton-hour while EPRI (1992) reports a range of $21 - $50 per ton-year. The following equation (Cler 1995) can be used to estimate the annual maintenance cost for an absorption chiller:

$$Cost(\$) = (EFLH)(tons)(MC) \qquad \textbf{Eq 29}$$

In Equation 29, EFLH is the equivalent full load hours of operation, tons is the rated capacity of the chiller, and MC is the maintenance cost in dollars per ton-hour. The following analysis looks at the LCC comparison between a double-effect, direct-fired absorption system and an electric motor-driven centrifugal chiller that was given in EPRI (1992) and evaluates it from the standpoint of the cost components discussed above. Table 19 provides the basic system operating assumptions used in the analysis and comparison. If the EFLH value in Table 14 is used in the relation:

$$EFLH = (24)(CDD)/(\Delta T_0) \qquad \textbf{Eq 30}$$

Table 19. System operating assumptions.

Parameter	Vapor Compression	2-Effect Absorption
*Ton Rating	900	900
*Operation (hr./yr.)	2300	2300
ΔT_o	50°	50°
*EFLH (hr.)	1277	1277
Equivalent CDD/yr.	2660	2660
Purchase Cost ($)	270,000	578,000
*COP	5.0 (632 kW Drive)	1.0
*Efficiency Factor	0.784	1.0
*Maintenance ($/ton)	18	24
*Water Treatment ($/ton)	18	27
*Demand Charge ($/kW)	100	100
* Note: Values obtained from EPRI (1992)		

One can solve for CDD and find that the result for a ΔT_0 of 50° yields a CDD value of 2,660, which is very close to the value of CIDV for Chicago, IL (2,741) shown in Table 12. Table 20 lists the results of the cost component comparison. The values for the right hand column in Table 20 were taken from the EPRI (1992) report. The values in the left hand column were obtained using relations developed in this study as well as some of the assumptions from the EPRI report. The demand charge was obtained by multiplying the demand charge rate by the kW rating of the compressor drive. The energy charge is found by using Equation 30 to estimate the total electric usage for the year of 1,030,641 kWh and then multiplying by the electric charge of $0.045/kWh. The EPRI report uses an equivalent electric charge of $0.106/kWh. This also takes into account the energy cost for the system auxiliaries. The energy required for auxiliaries is assumed to be about 15 percent of the chiller energy requirements. For this study, that translates into 154,596 kWh = (0.15)(1,030,641). Multiplying this result by the energy cost of $0.045/kWh yields $6,957. The dollar per ton costs for maintenance were taken from the EPRI report. However, the cost analysis in the EPRI report did not include the water treatment costs.

The two methods provide relatively the same results for the total annual operating costs. Table 21 shows the comparison of the cost components from the EPRI report and this study for a double-effect, direct-fired absorption system. Again, the right hand column contains the values from the EPRI report and the left hand column contains values obtained from equations developed in this study as well as applying some of the assumptions from the EPRI report. The total annual natural gas requirements were estimated at 13,789 MMBtu (million Btu) from the following equation:

$$MMBtu / yr = (ton)(CDD)(288{,}000) / (10^6)(COP)(\Delta T_0) \qquad \textbf{Eq 31}$$

The constant 288,000 is a conversion factor for tons to Btu. The gas cost for this study was determined using a rate of $3.5/MMBtu. Gas costs given in $/therm can be converted to $/MMBtu by multiplying by 10. The auxiliary electric requirement of 169 kW was found by using the ton rating and the electric use

Table 20. Operating cost comparison, vapor compression.

Parameter	Study Results	EPRI Report (pp 11-10)
Demand Charge	$63,200	$62,280
Energy, Chiller	$46,379	$60,978
Demand Charge, Aux.	$8,600	$19,044
Energy, Aux.	$6,957	Included in Energy, Chiller
Total Energy	$125,136	$142,302
Maintenance	$16,200	$16,200
Water & Treatment	$16,200	Not Included
Total Operating Cost	$157,536	$158,563

Table 21. Operating cost comparison, 2-effect, absorption system.

Parameter	Study Results	EPRI Report (pp 11-10)
Gas Cost	$48,263	$46,242
Aux.Electric Charge	$9,711	$38,916
Aux. Demand Charge	$16,900	$16,920
Total Energy Cost	$74,876	$102,078
Maintenance	$21,600	$5,400
Water & Treatment	$24,300	$25,881
TotalOperating Cost	$120,775	$132,682
Operating Savings (difference between Tables 14 and 15)	$36,761	$25,881

per ton for chiller, fans, and pumps in the system. An electric charge rate of $0.045/ kWh was used to obtain the electric charge. The demand charge was determined using the auxiliary electric requirement and a $100/kW-yr demand charge. Maintenance costs were determined using dollar per ton costs from the EPRI report. However, we note a discrepancy in the EPRI calculation for the maintenance cost. At $24/ton for a 900-ton chiller, the maintenance cost should be estimated at $21,600 rather than $5,400. The resulting total annual operating costs are relatively consistent (within 10 percent) for the two methods compared. The operational savings of the absorption system over the vapor compression system is shown in the last row of Table 21.

The difference in purchase costs between the vapor compression and absorption system are ($578,000 - $270,000) = $308,000. Table 22 shows the LCC comparison between the vapor compression and absorption system using the operating costs developed in Tables 20 and 21. Since the absorption unit has lower operating costs than the electrical unit, we can look at the simple payback period. This is found by dividing the difference in purchase costs by the operational savings. The EPRI analysis shows a payback in 11.9 years while the results obtained in this study show a payback in 8.4 years. Simple payback is often used to get a rough feel for the benefits of one option over another. However, this method does not provide a good basis for financial decisionmaking since it does not take into account all the cash flows or the time value of money. The discounted payback takes into account the time value of money, but does not take into account all the cash flows. For this analysis, the discounted payback is between 10 and 11 years, while for the EPRI analysis, the discounted payback is beyond the 15-year assumed life of this project.

Table 22. LCC Comparison.

Parameter	Study Results	EPRI Report
Simple Payback	8.4 years	11.9 years
Discounted Payback	10 to 11 years	> 15 years
NPV Absorption	-$1,920,825	-$2,053,212
NPV Vapor Compression	-$2,021,548	-$2,032,967

6.6.2 NPV

In this study, the absorption system can be shown to be the better choice of the two systems by a margin of about $100,000 in NPV. The EPRI results show that the vapor compression system is the better choice, but only by about $20,000 in NPV. For the EPRI results, the two systems are essentially even while this study shows a more distinct advantage for the absorption system. The results of this type of comparison can depend a lot on the assumption made in determining the cash flows and their timing. The choice of a discount rate can also have a big impact on the results. Since the vapor compression system requires higher cash outflows for operation, a higher discount rate will cause the NPV of the two system to be closer. If we used a discount rate of 8 percent (LCCM = 8.5594) instead of 4 percent, the NPV of the two systems based on this study would be essentially the same (approximately $1.6 million).

One factor not considered in the NPV calculation of Table 22 is the effect of depreciation. For commercial firms, the depreciation of capital equipment is considered a periodic expense and can be used to offset any income for that period. This reduces taxable income and provides what is called a "tax shield." By using a straight line depreciation for each system, one obtains a yearly depreciation of $38,533 for the absorption system and $20,000 for the vapor-compression system. If the tax rate is 40 percent, the absorption system will provide a tax shield of (0.4)($38,533) = $15,413 and the vapor compression system will provide a shield of $80,000. This is considered a cash inflow since the depreciation leads to paying out a reduced tax by the shield amount. The higher purchase cost of the absorption system will provide a higher tax shield than the vapor compression system over the life of the project. Table 23 shows the break-even gas-electric price for given gas prices based on the analysis in this study.

Table 23. Break even energy prices.

	Gas Cost ($4.15/MMBtu)	Gas Cost ($3.5/MMBtu)
Electric Cost	$0.045/kWh	$0.035/kWh

7 Discussion

The procedures for estimating life cycle costs have been outlined in the reports of EPRI (1992), SAI Corp. (1995), WGNAS (1993), and Cler (1995). The objective of this study was broader than simply formulating another set of cost estimates; this study undertook to compare procedures and results that may lead to a common basis for comparison. In doing so, this study has presented the whole matrix of refrigeration and heat pump systems, capacities, energy sources, applications, and price ranges. While this review may appear critical in sorting out discrepancies and their sources, performing a critical comparison was not the aim of this study.

The procedure began with identifying a few candidates among the ones listed in the matrix. The selection was made for a given capacity at a given location, its degree days, requirement for temperature and humidity of the space under consideration, energy sources and costs (electric or gas), available heat sink, and source. Comparisons were made on the savings and life cycle cost (LCC), the cost of owning and operating, and the net present value (NPV) based on standard commercial accounting practices, although the practice of paying for a government project and its first cost up front, and the need to be concerned about the subsequent yearly operating expenses may qualify these practices. The choice may be influenced by the budgetary climate at a given time. Cost projection is influenced by future prices of electricity and natural gas, estimated by EIA of DOE (EPRI 1992) to have the averages listed in Table 24.

Uncertainties are perturbation by import of LNG, interest rates, and pollution control legislation on electric cost. Deregulation and futures trading of electricity (Coy 1996) will lead to increased competition and price fluctuations.

Criteria were identified for unique applications that will determine the choice of equipment and eliminate many from consideration. All possible energy sources were also identified, including a solar pond. The latter was determined to be uneconomical in terms of $/Btu in comparison to natural gas.

Table 24. Estimated future electricity and natural gas prices.

Year	Electricity, $/kWh	Natural gas. $/MMBtu
1991	0.070	4.90
2000	0.073	4.30

When engine-driven chillers are used, their advantage is in functioning also as heat pumps and in recovery of exhaust heat. However, for a location such as WGNAS where winter ambient temperatures may reach 10 °F or lower, the peak heating demand in the winter may almost double (0.4 MMBtu/hr for the facility under consideration) the demand that can be produced by a heat pump adapted from the chiller (0.24 MMBtu/hr); economy may call for the use of additional furnace heating rather than increasing the capacity of the heat pump/chiller system. When used in family housing, noise of 60 to 72 dB can become a concern. Acceptance of running 5-ton engine-driven chiller/heat pump units all night long along rows of houses needs careful consideration. Moreover, the requirement for engines of 20 to 100 hp to have 15-year life in continuous service calls for a longevity that exceeds the best of automobile engines (1,000,000 miles). In the low power range, Stirling engines may deserve a further look for their efficiency and quiet operation. Centrifugal units of large tonnage, say, 1,000-ton level, may be driven with gas turbines with reducing gears, but gas turbines of compatible capacity will have lower efficiency and will need more frequent replacement than the gas engines. The latter, with gas injection, has a favorable heat rate of 7,000 Btu/hp hr (36.5 percent thermal efficiency) or lower, and can satisfy the long life requirement. Using step-up gears to drive a centrifugal compressor is no problem (a practice since the 1950s in natural gas pumping along transmission pipelines). This study takes exception with the SAI (1995) report's comment that gas engine-driven centrifugal units produce "undesirable vibrations."

Absorption units have not been popular at Army bases. They need technicians trained with vacuum technology (LiBr-H_2O system operates at 6.5 mmhg pressure) (see "Maintenance Needs," Section 5.2.2, p 52), although it should not be an important obstacle. When a large, gas-fired absorption unit is deemed favorable, the competition will be a steam absorption system along with a cogeneration plant. The latter will allow the flexibility of a variety of air-conditioning systems, such as a special system for a base hospital. A study on the planning for chiller/heat pump installations should include cogeneration and its retrofitting as an option. Still another alternative might be to use the standby power plant at a base for peaking.

The operation cost of chillers at Army installations remains to be validated. Cost analysis that includes operation of chiller/heat pump combinations remains to be done. With Congressional funding for demonstration of natural gas cooling systems, 43 DOD installations are either operating or installing engine-driven chillers, absorption chillers, and desiccant cooling systems, as of early 1997 (Sohn et al. 1997). Actual construction and operation cost will be collected from these projects and compared with the cost data documented in this report.

Collection and analysis of energy efficiencies of those systems in field operation will yield a valuable guideline for improvement of the life cycle cost analysis. These steps are recommended for further study.

8 Conclusions

Gas engine driven vapor compression systems represent a feasible alternative for military installations in some situations. This technology is appropriate for small cooling loads (below 20 tons) when excessive electrical demand charges can be avoided by the use of inexpensive natural gas. Therefore, the use of this technology depends on the nature of the electric utility rate structure. Potential applications include barracks, clubs, and mess halls. Additional efficiency can be obtained by using waste heat from the exhaust. One problem that remains to be overcome in these applications is excessive engine noise and durability for extended periods of use.

Steam powered absorption systems suffer from low COP values, high initial system cost, and higher annual maintenance costs. Therefore, military installations may have limited use for these systems. However, steam-powered absorption systems are unique in their ability to use waste heat. As the economic analysis of absorption systems illustrated, these systems can provide an economic benefit when used in conjunction with cogeneration facilities and sources of waste heat. Installations currently serviced by electric utilities that include relatively high demand charges in their rate structure may reap additional economic benefits.

Thermoelectric coolers may be considered for cooling applications on the scale of fractional ton sizes. Due to the expense and low efficiencies of such systems, no installation-wide applications for the military currently exist. Still, the use of this technology will remain important in military applications for the cooling of electronic equipment and sensors.

Desiccant systems are applicable in Army situations that require very dry environments. A desiccant system is useful only where humidity below 36 gr/lb dry air or 0.005 kg/kg dry air is required, or where drying without cooling is needed. Applications include the storage of materials, particularly machinery and armaments, to prevent deterioration due to corrosion. Note that standard vapor compression systems are the more economical choice for higher air temperatures and moisture conditions.

The use of heat pumps in combination with geothermal sources and sinks has the potential to provide efficient cooling for military housing. Continued investigation into this possibility is recommended. Experience with a geothermal heat source or sink has been limited to small unit capacity systems of 5 to 25 tons, in which a bore hole is used for the heat source or sink, adding $600 to $1,000 per ton to the installation cost. The use of geothermal heat pumps is feasible for mild climates, such as in the southern United States. The higher winter temperatures provide a thermodynamically superior operating condition than the extreme cold in Northern climates. Fort Polk, LA, recently installed ground-coupled heat pumps in 4,003 family housing units as part of a shared energy savings contract. Preliminary results of this project are positive. The use of surface water in such applications may offer economic advantages, but it has its own unique problems. Users of lake water or water from aquifers are reminded of the average rate of 2.5 gpm/ton of water flow. Extensive usage at a given location may cause significant environmental problems. Careful consideration is called for if groundwater is to be returned to the aquifer after passing through the chiller system.

The use of a salt water solar pond as an energy source for an absorption system tends to be costly within the latitudes of the United States and is not recommended as a cost saving alternative.

The baseline case electric motor driven vapor compression system has proven to be reliable and economical in military installation applications. The main drawback of an electric drive is the potential for a high electric demand charge. The latter can be eliminated by using: chilled water/ice storage, or cogeneration (which is desirable in a large base facility). The alternative refrigeration systems outlined in this study have the potential to provide long term cost savings if they are used in appropriate circumstances.

In all cases, the results of a careful evaluation of the needs and characteristics of each installation should be the primary consideration when evaluating alternative cooling systems. In all these situations, decisionmaking can be assisted by determining the first cost and LCC as a basis for a valid comparison among alternatives on the basis of NPV.

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Abbreviations and Initialisms

$(COP_c)_e$	Coefficient of performance of a refrigerator based on mechanical energy input
$(COP_c)_s$	Coefficient of performance of a refrigerator based on heat input
$(COP_h)_e$	Coefficient of performance of a heat pump based on mechanical energy input
$(COP_h)_s$	Coefficient of performance of a heat pump based on heat input
M	Mass flow rate of exhaust gas
N_c	Cooling capacity in ton of refrigeration
N_{hp}	Heat pump capacity in tons
Q_c	Heat removed from the chilled water
Q_h	Heat delivered for heating
Q_o	Heat removed from the surroundings
Q_s	Heat supplied from a heat source to drive the system
T_c	Temperature of chilled water
T_h	Temperature at which heat is delivered for heating by a heat pump
T_o	Temperature of the surroundings or heat sink (for the condenser of a refrigerator) or source (for the evaporator of a heat pump)
T_s	Temperature of the heat source to drive the system
W	Work input from a shaft (or electric motor or an engine)

η_e	Thermal efficiency of engine driving a refrigerator or thermal efficiency of motor based on generating efficiency (summer generating or engine efficiency)
η_e'	Thermal efficiency of engine driving a heat pump or thermal efficiency of motor based on generating efficiency (winter generating or engine efficiency)
ARI	Air Conditioning and Refrigeration Institute
CCD	Charge coupled device
CDD	Cooling degree days
CDHR	Hourly cooling degree days
CDHRV	Ventilated hourly cooling degree days
CFC	Chlorofluorocarbon
CID	Cooling insolation days
CIDV	Vented cooling insolation days
CLGC	Closed-loop ground coupled
db	Dry bulb temperature
DCC	Double-condenser coupled
DOE	Department of Energy
DX	Direct expansion
EER	Energy efficiency ratio
EFLH	Equivalent full load hours of operation
EIA	Energy Information Administration
EPRI	Electric Power Research Institute

GCHP	Ground coupled heat pump
GHP	Gas engine heat pump
GSHP	Ground source heat pump
GWP	Global warming potential
HCFC	Hydrochlorofluorocarbon
HDD	Heating degree days
HDHR	Hourly heating degree days
HHV	Higher heat value
HID	Heating insolation days
HVAC	Heating Ventilation and Air Conditioning
LCC	Life cycle cost
LCCM	LCC multiplier for given life and discount rate
LF	Load factor
LHV	Lower heating value
MC	Maintenance cost
MITI	Ministry of International Trade and Industry (Japan)
NCG	Non-condensable gas
NOAA	National Oceanic and Atmospheric Administration
NPV	Net present value
OLGW	Open-loop ground water
RT	Refrigeration ton

TEC	Thermoelectric cooler
UA	Overall heat transfer coefficient
wb	Wet bulb temperature
WGNAS	Willow Grove Naval Air Station
WLHP	Water loop heat pump

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